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*16. Abstract* The purpose of the program was the construction and hydraulic operation of flow-work exchangers in order to test the performance of each component part and to pinpoint problems which have to be taken into consideration in later designs. In the reverse osmosis process, the feed system must be pressurized to the operating pressure (400 to 1500 psi) and the reject brine has to be depressurized before being removed from the system. A flow-work exchanger is a unified piece of equipment which simultaneously pressurizes a condensed fluid stream and depressurizes a substantially equivalent volume of another condensed fluid stream. After a brief testing of a preliminary flow-work exchanger unit, two working units were constructed. The first unit, which has two floating-piston type displacement vessels, is operable under 1500 psig and delivers 9 gpm. The second unit, which has two bladder-type displacement vessels, is operable under 1500 psig and delivers 18 gpm. The initial study was limited to the construction and hydraulic operation of these units without coupling them to a reverse osmosis unit. The overall efficiencies of the units were as high as 90 to 95% at the rated capacities.

*17a. Descriptors*

Energy Transfer, Desalination Apparatus

*17b. Identifiers*

Flow-Work Exchanger, Condensed Fluid Stream, Floating Piston Type, Bladder Type

*17c. COWRR Field & Group*

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# A Flow Work Exchanger for Desalination Processes

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Contract No. 14-01-0001-1166

UNITED STATES DEPARTMENT OF THE INTERIOR • Stewart L. Udall, Secretary
Max N. Edwards, Assistant Secretary for Water Pollution Control

[redacted]

Created in 1849, the Department of the Interior—America's Department of Natural Resources—is concerned with the management, conservation, and development of the Nation's water, wildlife, mineral, forest, and park recreational resources. It also has major responsibilities for Indian and Territorial affairs.

As the Nation's principal conservation agency, the Department of the Interior works to assure that nonrenewable resources are developed and used wisely, that park and recreational resources are conserved for the future, and that renewable resources make their full contribution to the progress, prosperity, and security of the United States—now and in the future.

## FOREWORD

This is one of a continuing series of reports designed to present accounts of progress in saline water conversion and the economics of its application. Such data are expected to contribute to the long-range development of economical processes applicable to low-cost demineralization of sea and other saline water.

Except for minor editing, the data herein are as contained in a report submitted by the contractor. The data and conclusions given in the report are essentially those of the contractor and are not necessarily endorsed by the Department of the Interior.

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## SUMMARY

In the reverse-osmosis process, the feed stream must be pressurized to the operating pressure (400 to 1500 psi) and the product water issues from the unit at essentially atmospheric pressure. The reject brine stream remains at the high pressure and has to be depressurized before being removed from the system. Therefore, we have a problem of simultaneously pressurizing a fluid stream and depressurizing another fluid stream. Power recovered from the reject brine stream would be used to pressurize some of the incoming feed stream to reduce the overall energy requirement of the process.

A flow-work exchanger is a unified piece of equipment which simultaneously pressurizes a condensed fluid stream and depressurizes a substantially equivalent volume of another condensed fluid stream. A flow-work exchanger uses two displacement vessels to form closed loops with a high pressure processing system. Each of the displacement vessels is alternately filled by a low-pressure feed and a high-pressure product, both pressurized and depressurized, respectively, by substantially non-flow processes. The pressurized feed is pushed into the processing system by the high-pressure product stream and the depressurized product stream is pushed out of the displacement vessel by the low-pressure feed stream. A paper describing flow-work exchangers has been published [see A.I.Ch.E. Journal 13, No. 3, 438-442 (1967)].

Under contract 14-01-0001-1166, a preliminary testing unit was first constructed by a local machine shop in order to test the performance of each component part and to pinpoint problems which have to be taken into considerations

in later designs. After a brief testing of the preliminary unit, two working units have been constructed. The first unit, which has two floating-piston type displacement vessels, is operable under 1500 psig and delivers 9 gpm. The second unit, which has two bladder-type displacement vessels, is operable under 1500 psig and delivers 18 gpm. Most of the component parts used in these units were adopted from commercial products with some modifications. This initial study has been limited to the construction and hydraulic operation of the units without coupling them to a reverse-osmosis unit. The hydraulic operation of these units has been generally satisfactory and the technical feasibility of such units has been demonstrated. The overall efficiencies of the units are as high as 90% to 95% at the rated capacities. These units are described in Sections V and VI of Part B.

Based on the observation of the operation of these units, improvements to be incorporated in future units have been suggested in Section VII, Part B. These improvements will simplify the construction of a unit, simplify its start-up operation, and, possibly, give rise to smoother operations. Modifications required for coupling a flow work exchanger unit to a reverse-osmosis unit have also been suggested.

Some efforts have been made in the prior art search to find alternative ways of recovering energy from the high pressure brine stream. The results of the study are summarized in Part A.

## Part A. REVIEW OF CONVENTIONAL POWER RECOVERY SCHEMES.

### I. INTRODUCTION

Whenever a high pressure fluid stream is depressurized in such a manner that throttling occurs, there is a potential for power recovery. Power recovery from high pressure gas streams has been rather commonly practiced in process industries such as air separation plants, catalytic-cracking plants, ethylene recovery plants, and waste treatment plants using the Zimmermann process of wet oxidation of sewage sludge (1, 2, 3, 4, 5, 6). However, the problem of power recovery from high pressure liquid streams has been more or less neglected up to recent years. The most promising applications would be in power recovery from relatively large liquid streams undergoing medium to high pressure drops. It has been reported that power has been successfully recovered from liquid streams both in hydrocracking plants and natural gas processing plants (1, 7, 8, 9, 10). In the hydrocracking plants, the feed stocks are pressurized to the 1,500-2,000 psi operating pressure and the discharge streams are depressurized through multistage centrifugal machines. In the natural gas processing plants, the rich oil streams leaving the bottom of high pressure gas absorbers are reduced in pressure at differentials of 250 to 550 psi through rotary positive-displacement machines. Power recovered in these depressurization operations is utilized in pumping operations.

In the conventional schemes, the power recovery realized from liquid streams is usually in the form of shaft work. The power recovery machine can be either of the reciprocating type or of the rotary type. The rotary machine can be a centrifugal-type, an impulse turbine type or a positive displacement type. Both theoretically and practically, any pump can be run backwards to serve as a power recovery machine. These power recovery machines are separately described below:

## II. CENTRIFUGAL-TYPE UNIT

The centrifugal-type power recovery machine is basically a centrifugal pump running backwards. For a power recovery application, the high pressure liquid enters the machine at what normally would be the discharge nozzle of a pump. The impellers rotate in the opposite direction as compared to a pump allication (7, 11, 12). It has been stated that a good centrifugal pump operating with high efficiency may be expected to display good performance when the direction of flow is reversed and the pump is used as a driver. It has farther been stated that the efficiencies of a centrifugal-type machine used both as a pump and as a power-recovery machine may be considered identical with little error at their respective best efficiency points and at the same speed (12). The performance characteristics of typical centrifugal-type machines used both as pumps and power-recovery machines have been presented by C. P. Kittredge (13). The hydraulic design of centrifugal pumps and water turbines has been described by H. H. Anderson (14). Efficiency of a centrifugal-type unit varys greatly with flow-rate and speed; extremely low efficiencies occur for low-flow, low-speed conditions. Therefore, a centrifugal type unit should be operated near its design speed and flow rate (9). For estimation purposes, the efficiency may be assumed to be on the order of 80%. A centrifugal unit is less sensitive to non-lubricating properties of the fluid and foreign materials suspended in the fluid.

J. W. Purcell and M. W. Beard (7) have described in detail the power recovery schemes used in hydro-cracking plants. A typical unit is a six-stage diffuser type unit with barrel casings. Multistage units are used in these plants, because of the large pressure differential (1500-2000 psi) involved. In a typical installation, a six-stage unit is used to drive a 12-stage feed pump. A motor with a double extended shaft is mounted between the pump and the power recovery unit to supply the make-up power. This type of arrangement is referred to as a tandem arrangement. Experience gained in this field is of particular interest, because this power recovery scheme can be used in reverse osmosis systems. D. T. Bray and H. F. Menzel (15) recommended the use of a recovery turbine to drive a generator and utilize the electric power recovered to drive a multistage centrifugal pump, such as a Byron Jackson Model DVMS pump.

## III. HYDRAULIC WATER TURBINE UNIT

A hydraulic water turbine unit has been developed mainly for hydro-electric applications, and it is not suitable for most process applications. The major manufacturers of hydraulic water turbines have so far made no effort to sell their products to the process industries. Allis-Chalmers Co., Baldwin-Lima-Hamilton Corp., the James Leffel & Co., and Dominion Engineering Co. have been contacted for their comments on the feasibility of applying a hydraulic water turbine unit in the power-recovery in a reverse osmosis system. Most of the companies have agreed that it is feasible to use an impulse turbine in the power recovery. It is possible either to couple the impulse turbine to a generator or to couple the impulse turbine directly to the pumping unit being used to pressurize the feed brine. Since the pump requires more power than that produced by the turbine, a motor will be required to supply the balance.

In general, a large, vertical, multi-nozzle impulse turbine would operate at a peak efficiency between 88% and 92%. The most important factor affecting the economic justification of this type of power recovery is the available flow. It has been stated by Allis-Chalmers Co. that for the minimum operating pressure of 400 psi, a minimum flow of 500 gpm would be required to keep the cost of the turbine within practical limit.

The Bureau of Reclamation has published a useful monograph describing the selection of hydraulic reaction turbines (16). N. N. Kovalev's book on "Hydroturbines" is also of great help (17).

## IV. ROTARY POSITIVE-DISPLACEMENT TYPE UNIT

The rotary positive-displacement type power recovery machine is basically a rotary positive-displacement pump running backwards. Such a unit will be simply referred to as a hydraulic motor when it is used for power-recovery and a hydraulic pump when it is used for pumping. Recently, appreciable reductions in the price, weight, and size of hydraulic motors and hydraulic pumps have been made (18, 19). Today one can buy a 20-hp fluid motor to fit inside an envelope of about 0.2 $ft^3$. A 20-hp electric motor is twelve times as large as the same size fluid motor (18). The positive-displacement type units will generally be lighter in weight and take less space than the centrifugal type units (9). A rotary positive-displacement type unit is a close-clearance unit which is very sensitive to the non-lubricating properties of the fluid handled and to suspended foreign matter. As has been described, machines of this type have been used in the power recovery of rich oil from gas absorbers in natural gas processing plants. A machine of this type run with a fluid of good lubricity and reasonable viscosity has a very high efficiency. Therefore, it is very unfortunate that a machine of this type cannot be applied in water hydraulics due primarily to the non-lubricity of water and brine.

The power recovery problem in a reverse osmosis system is to recover power during the depressurization of the reject brine and utilize the power to pressurize some of the incoming feed brine stream. The reverse situation is the case in the so-called "hydrostatic drive" in oil hydraulics. In the hydrostatic drive, a hydraulic pump is used to pressurize a hydraulic fluid and the high pressure hydraulic fluid is used to drive a hydraulic motor. It has been claimed that the overall efficiency of such a system can be as high as 90%. A very good discussion on hydrostatic drive is given by Werner Holzbock

(20). Had it not been for to the non-lubricity of water, similar equipment could have been used in a reverse osmosis system and a comparable overall efficiency could have been obtained.

Hydraulic motors and hydraulic pumps have been classified into the following 17 types:

i. Vane Pumps - a. Vanes-in-rotor pump, b. Vanes-in-stator pump, c. Slipper-vane pump, d. Flexible-vane pump.

ii. Gear Pumps - a. Spur-gear pump, b. Internal-gear pump, c. Progressing-tooth gear pump.

iii. Piston Pumps - a. Axial piston pump, b. Radial piston pump, c. Eccentric-ring plunger pump, d. Intensifier-piston pump.

iv. Screw Pumps - a. Multiple screw pump, b. Single screw pump.

v. Lobe Pumps

vi. Diaphragm Pumps

vii. Squeegee Pumps - a. Flexible line pump, b. Flexible tube pump.

Warren E. Wilson (18) has given the most interesting display of the effects of kinematic viscosity of the working medium and pump speed on the volumetric efficiency and the overall efficiency of gear pumps, vane pumps, and piston pumps. The graphs prepared by him clearly show how the overall efficiencies of the units vary with kinetmatic viscosity and the pump speed. The efficiency is low for a highly viscous fluid because of the high friction, and is low for a low viscosity fluid because of the high slip. Therefore, the difference between oil hydraulics and water hydraulics has been clearly displayed in this work.

Several companies, Bellows-Valvair Co., Racine Hydraulics & Machinery Inc., Air Equipment Co., Tektron Co., The Weatherhead Co., and Be-Ge Manufacturing Co., have been contacted for their comments on the feasibility of applying the hydraulic machines of their manufacture in the power-recovery of a reverse osmosis system. Their comments have generally been negative. It should be mentioned in this connection, however, that Hypo. Inc., Minneapolis, Minn. has been developing compact radial plunger type machines with some success. A breakthrough in overcoming the non-lubricity problem is still needed.

## V. RECIPROCATING TYPE UNIT

A positive displacement type machine can usually be used either as a pressurizer or a depressurizer with minor modification. Therefore, inquiries were sent to manufacturers of high pressure reciprocating pumps asking for the feasibilities of using a high pressure reciprocating pump driven by a high pressure fluid (in the place of a high pressure air drive, a high pressure steam drive or a power drive).

Crane Co. replied that power recovery of this type is practical and is currently practiced in the petroleum field. The company also suggested that Ingersoll-Rand Co. and Pacific Pump Co. should be contacted, since they were actively engaged with systems of this type. The Platt Manufacturing Corporation, New York City, indicates that a hydraulically operated reciprocating pumping unit for power recovery is definitely feasible, but that the only units of this type built today are not for the commercial market, their use being of a restricted nature by the large chemical manufacturers. The building of such a pump was indicated to be a costly development. Based on information received from the Crane Co. and Platt Manufacturing Corporation, overall efficiencies for this type of system were estimated to range from 70 to 85%.

Other companies contacted are Ajax Iron Works, Corry, Pa., Dean Brothers Pumps, Inc., Indianapolis, Ind., Logemann Bros., Co., Milwaukee, Wis., and Worthington Corp., Harrison, N. J.

## VI. SUMMARY

From the review of the prior art search described above, it may be summarized that the following three schemes can be used for the power recovery in a reverse osmosis system:

1. We may use a multistage centrifugal machine to extract power from the reject brine and use the power either to drive a generator or multistage centrifugal feed pump. The overall efficiency will be on the order of 60%.

2. We may use an impulse type hydraulic water turbine to extract power from the reject brine and use the power either to drive a generator or a multistage centrifugal feed pump. The overall efficiency is estimated to be less than 70%.

3. We may use a hydraulically driven reciprocating pump. The overall efficiency is estimated to be less than 80%.

Power recovered in case 1 and case 2 may also be used to drive a plunger pump. Then, a mechanism is required to convert the rotary motion into the reciprocating motion.

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Part B. FLOW WORK EXCHANGER

I. INTRODUCTION

The power recovery unit in a reverse osmosis system is for recovering power from the high pressure reject brine and utilizing the power so recovered in pressurizing some of the incoming brine. The conventional power recovery schemes have been reviewed in Part A. When such conventional schemes are used, we need a power recovery unit for depressurizing the reject brine and a high pressure pump for pressurizing the feed. Therefore, we need a set of machines consisting of a depressurizer (a power recovery unit) and a pressurizer (a high pressure pump). The overall efficiency of the combined unit, defined as the fraction of energy theoretically recoverable from the high pressure reject brine which is actually absorbed by the incoming feed brine, is less than 60%, 70% and 80% respectively for scheme 1, scheme 2 and scheme 3 described in the summary of Part A. Rather large amounts of shaft work are transmitted through these machines. Therefore, large machine members and driving mechanisms are required and rather elaborate finishing and packing are required in manufacture. These lead to high equipment cost.

A flow-work exchanger herein described is a unified depressurizer-pressurizer unit which applies to a simultaneous pressurization of a condensed fluid A and depressurization of equivalent volume of another condensed fluid B. The fluids, A and B, are pressurized and depressurized respectively, by substantially non-flow processes. The movements of the fluids are conducted against small pressure differentials and thus $\Delta(PV)$ values for the movements of fluids are small and the shaft work is greatly reduced. This scheme of flow work exchanger was first introduced by Cheng and Cheng in connection with their "Freezing Process which is Based on the High Pressure Inversion in the

the Order of Melting Points," (1). A paper describing this power recovery scheme has been published by Cheng, Cheng, and Fan (2). The optimal operating condition of a high-pressure process is greatly influenced by the efficiencies of the pressurization and depressurization operations. In addition to the immediate cost reduction obtainable by the adoption of flow work exchangers in a process where the operating conditions is substantially left unaltered, cost reduction can also be realized by operating the process under the new optimal operating condition. This further cost reduction may be significant in many cases. Fan, et al. have made an optimization study to show the effect of improving the power-recovery efficiency on the economy and the optimal operating condition of a reverse osmosis system (3, 4).

Displacement vessels have been used by J. H. Watts (5), B. L. Mann (6), C. J. Ross (7), W. A. Swaney (8), L. E. Mills (9), H. L. Baer (10), and O. L. Nordin (11) for the purpose of introducing chemicals, additives, and solids into a pipe line. None, however, have applied them to power-recovery. The Warren Rupp Company (12) of Mansfield, Ohio has recently (1968) introduced the so-called "Dynaflex" slurry pump which combines the smoothness and dependability of a centrifugal pump with the abrasive and solids handling capabilities of a diaphragm pump. This Dynaflex pump utilizes some component parts which are used in the flow-work exchanger. Therefore, the experience gained in the development of this pump can be transferred to the development of the flow-work exchangers. A somewhat detailed description of this pump will be given as an Appendix for reference.

The purpose of the contract No. 14-01-0001-1166 is to construct working units in order to demonstrate the technical and economical feasibilities of the flow-work exchanger and to show its performance.

## II. THERMODYNAMICS OF A SIMULTANEOUS PRESSURIZATION-DEPRESSURIZATION PROCESS

Figure 1 illustrates a high-pressure processing system into which a feed is introduced by a pump $J_1$ and from which a product is discharged by a turbine T. The pump operates between pressures $(P_L)_1$ and $(P_H)_1$; the turbine operates between $(P_H)_2$ and $(P_L)_2$. Quantitative discussions will refer to a high-pressure process operated at 1,500 lb/sq in. gauge.

When a condensed fluid is pressurized without phase change to a high pressure, the reversible shaft work received by the fluid in a flow process $-w_f = \int Vdp$ is on the order of 200 times the corresponding value for a non-flow process $-w_{nf} = -\int pdV$. This is due to the noncompressibility of a condensed fluid; a liquid shrinks by about 1% upon the application of 100 atm. pressure. Similar statements can be made for the depressurization operation.

The reversible shaft work for the pump and turbine, shown in Figure 1, can be represented by

$$(-w_1)_f = \int_{(P_L)_1}^{(P_H)_1} Vdp$$

$$= [(P_H)_1 \cdot (V_H)_1 - (P_L)_1 (V_L)_1] - \int_{(P_L)_1}^{(P_H)_1} P \cdot dV \qquad (1)$$

$$(+w_2)_f = \int_{(P_L)_2}^{(P_H)_2} Vdp$$

$$= [(P_H)_2 \cdot (V_H)_2 - (P_L)_2 (V_L)_2] - \int_{(P_L)_2}^{(P_H)_2} P \cdot dV \qquad (2)$$

Each equation shows that the shaft work for a flow process is the sum of the shaft work for a corresponding nonflow process and the difference in the flow work terms under the high and the low pressure $\Delta(PV)$. They also show that

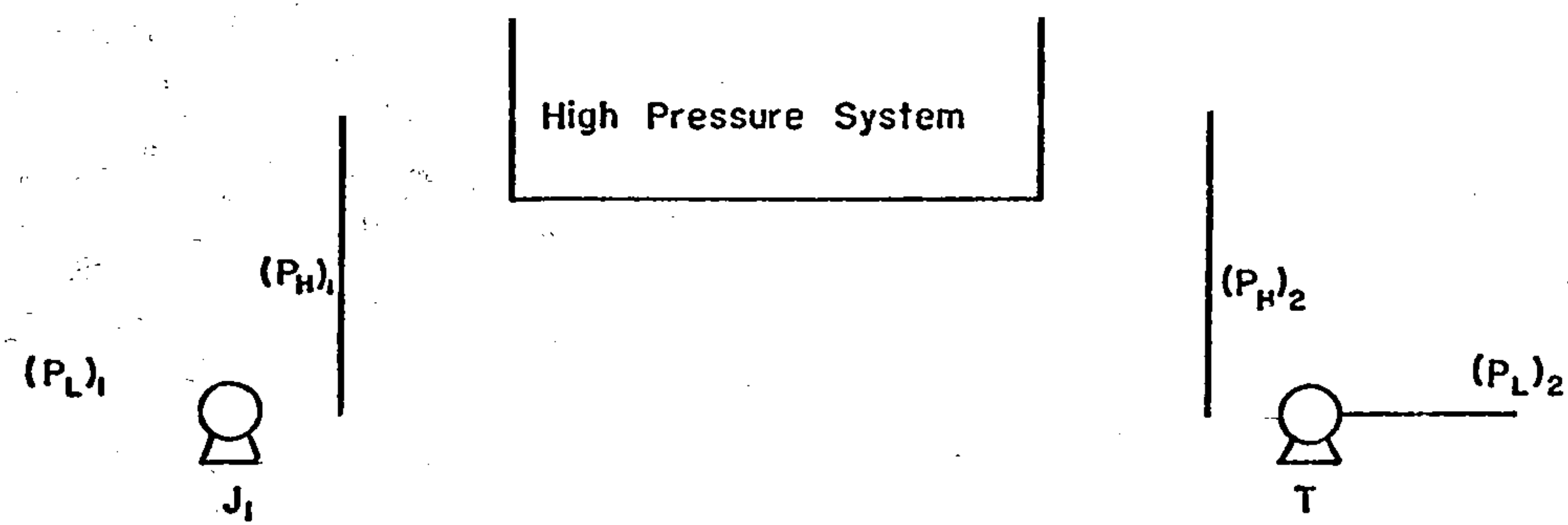



Fig. 1. A high pressure system with conventional pressurization and depressurization.

the large values of the shaft work for the reversible pressurization and the depressurization should be attributed to the large values of the differences in flow work terms $\Delta(PV)$'s.

A flow pressurization may be considered as a superposition of a nonflow pressurization and a movement of fluid. The $|\Delta(PV)|$ term is large, because the movement of fluid takes place across a large pressure differential between $(P_L)_1$ and $(P_H)_1$. Similarly, for flow depressurization the $|\Delta(PV)|$ term is large because of the movement across a large pressure differential between $(P_L)_2$ and $(P_H)_2$.

For flow pressurizing a condensed fluid and flow depressurizing another fluid simultaneously, as shown in Figure 1, it is possible to arrange the flow system so that movements of fluids take place across small pressure differentials; that is, between $(P_H)_1$ and $(P_H)_2$ and between $(P_L)_1$ and $(P_L)_2$. Then the $|\Delta(PV)|$ terms become very small, and the shaft work becomes small. The shaft work can approach the values of the corresponding nonflow processes.

The simultaneous flow pressurization and flow depressurization operation then involves the following steps: (1) low-pressure and small-pressure differential $(P_L)_1 - (P_L)_2$ displacement operation; (2) substantially nonflow pressurization of the feed; (3) high pressure and small pressure differential $(P_H)_2 - (P_H)_1$ displacement operation; (4) substantially nonflow depressurization of the product.

In the following discussions the first and third steps will be referred to as the low-pressure displacement operation and the high-pressure displacement operation, respectively. The entire operation will be called the flow-work exchange operation. A fluid to be pressurized in a process may exchange flow work with another fluid to be depressurized in the same process or in other processes. The equipment will be called the flow-work exchanger.

## III. ILLUSTRATION OF A FLOW-WORK EXCHANGER

Figure 2 illustrates a flow-work exchanger in connection with a high pressure reverse osmosis process. In a reverse osmosis process the feed stream must be pressurized to the operating pressure. The product water issues from the unit at essentially atmospheric pressure and the reject brine stream still remains at a high pressure and has to be depressurized. The flow-work exchanger illustrated in Fig. 2 is to recover the energy contained in the high pressure reject brine and utilize it to pressurize more feed brine. Since a flow-work exchanger exchanges flow-work between two condensed streams of essentially equivalent volume and the volume of the feed brine is greater than the volume of the reject brine, the excess portion of the feed brine has to be pressurized in a conventional way.

The figure shows that a flow-work exchanger consists of one or more displacement vessels (two, $0_1$ and $0_2$ as shown in Figure 2), check valves ($V_1$, $V_2$, $V_3$, and $V_4$), control valves ($V_5$, $V_6$, $V_7$, and $V_8$), a low-pressure low head pump ($J_1$), and a high-pressure low head pump ($J_2$). The pump $J_2$ is used to recover the pressure drop of fluid during its passage through the processing system and maintains $(P_H)_2$ higher than $(P_H)_1$ by an amount sufficient to carry out a high-pressure displacement operation to be described. Alternately, the pump $J_2$ may be installed at the inlet side of the high-pressure system such as location Y in Figure 2. The pump $J_1$ is used to maintain $(P_L)_1$ somewhat higher than $(P_L)_2$ to carry out a low-pressure displacement operation to be described. The high-pressure pump ($J_3$) is used to pressurize the excess part of the feed. The feed end and product end of a displacement vessel will be called the a end and b end, respectively.

Each displacement vessel is operated cyclically in the following steps.

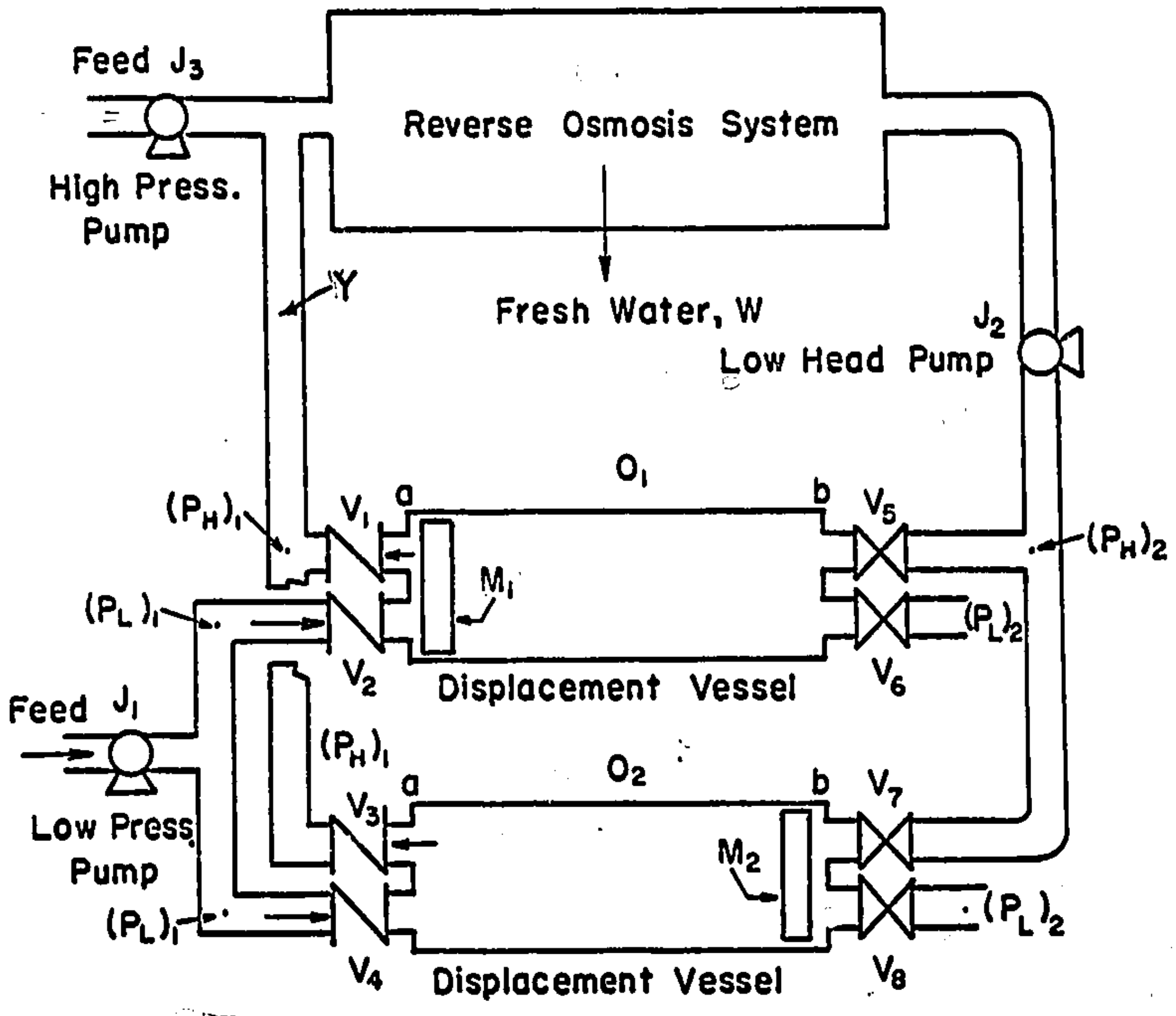



Fig.2 . A high pressure processing system incorporating flow work exchangers.

Step 1: Substantially nonflow depressurization. The displacement vessel $O_1$ is filled with the high-pressure product. By closing the valve $V_5$ and opening the valve $V_6$, the content in the displacement vessel is depressurized and some product fluid in the amount corresponding to the volume expansion due to the depressurization flows out of the vessel through the valve $V_6$. This operation takes a very short time. The check valves $V_1$ and $V_2$ are in the closed position during this operation.

Step 2: Low-pressure displacement operation. When the pressure in the vessel drops below $(P_L)_1$, the check valve $V_2$ opens and the low-pressure feed flows in through $V_2$ and the depressurized product flows out of the vessel through the valve $V_6$. The solid partitioner $M_1$ moves from the a end to the b end. The valves $V_1$ and $V_5$ are kept closed. At the end of this operation the vessel is filled with low-pressure feed.

Step 3: Substantially nonflow pressurization. The displacement vessel $O_2$ is now filled with the low-pressure feed. With the valve $V_8$ closed and the valve $V_7$ open, some high-pressure product flows into the vessel to pressurize the content. This operation takes a very short time, because only a small amount of fluid sufficient to compensate for the volume shrinkage has to be introduced. During this operation the check valves $V_3$ and $V_4$ are in the closed position.

Step 4: High-pressure displacement operation. When the pressure in the vessel exceeds $(P_H)_1$, the valve $V_3$ opens and the high-pressure product flows continually into the vessel through $V_7$ and the pressurized feed fluid is displaced into the high-pressure processing system. The solid partitioner $M_2$ moves from the b end to the a end. At the end of this operation the vessel is filled with high-pressure product. Then, it returns to step 1 and starts over again.

The displacement operations, steps 2 and 4, occupy most of the time in an operating cycle and each of the nonflow processes, step 1 and step 3, take rather short periods of time. Thus when two displacement vessels are operated with proper timing, fluid flow through the processing system will be continuous except for the short periods during steps 1 and 3 and the time taken in operating the valves. These disturbances may be lessened by accomodating a small accumulator in the system.

It may be noted that a three-way valve may be used to replace a pair of two-way valves, either $V_5$ and $V_6$ or $V_7$ and $V_8$. Furthermore a four-way valve may be used to replace the four valves, $V_5$ through $V_8$.

## IV. EFFICIENCY OF A FLOW-WORK EXCHANGER

The operation of a flow-work exchanger described in the last section can be illustrated by an idealized indicator diagram shown in Fig. 3 and the lost work involved in each step will be given later from the actual performance of the flow-work exchangers constructed under this project.

Referring to the figure, AB represents the volume of the displacement vessel $V_D$, AX represents the volume of the feed $V_F$ in the vessel, and BX represents the volume of the product $V_P$ in the vessel. Therefore

$$V_F + V_P = V_D \quad \text{and} \quad AX + BX = AB$$

The operational steps illustrated in the last section can be represented on the indicator diagram as follows.

1. Substantially nonflow depressurization. This operation is represented by 7-8 in Fig. 3. Point 7 represents the situation where the displacement vessel is filled with product fluid at pressure $(P_H)_2$. When the valve $V_6$ is open, the content is depressurized and its volume expands from 7 to 8, and its pressure drops to $(P_L)_2$. A volume of product fluid in the amount of 9-8 flows out of the vessel.

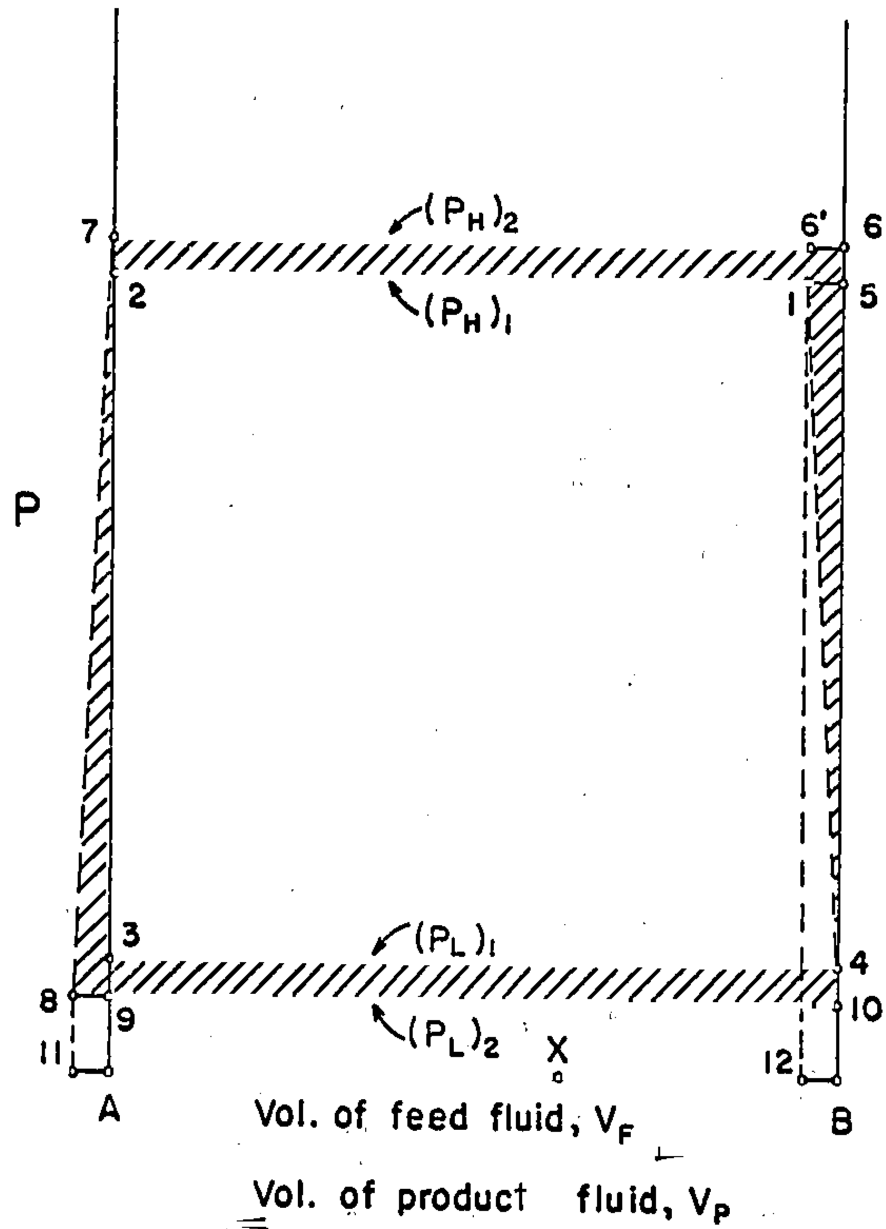



Fig. 3 . Indicator diagram of a flow work exchanger. (Schematic)

2. Low-pressure displacement operation. This operation is represented by $3 \rightarrow 4$ and $9 \rightarrow 10$ in Fig. 3. They show that feed fluid enters the vessel at $(P_L)_1$ and product fluid leaves the vessel at $(P_L)_2$.

3. Substantially nonflow pressurization. This operation is represented by $4 \rightarrow 1$ and $10 \rightarrow 6 \rightarrow 6'$. It shows that the high-pressure product is introduced into the vessel in the amount represented by 6-6' to compress the feed in the vessel to $(P_H)_1$.

4. High-pressure displacement operation. This operation is represented by $1 \rightarrow 2$ and $6' \rightarrow 7$ in Fig. 3. They show that the product fluid enters the vessel at $(P_H)_2$ and the feed leaves the vessel and is moved into the processing system at $(P_H)_1$.

The lost work involved in the steps is represented by the shaded areas. Area 7-8-9 is the work of expansion in step 1; it is so small that it may well be unrecovered to simplify the operation. Area 4-5-6-6'-1 is the irreversibility involved in step 3; and areas 3-9-10-4 and 7-2-1-6' represent the lost work involved in the displacement operation represented by step 2 and that represented by step 4 respectively.

The net work required in simultaneously pressurizing one fluid from $(P_L)_1$ to $(P_H)_1$ and depressurizing another fluid from $(P_H)_2$ to $(P_L)_2$ is zero in a reversible case. In an actual operation the net work to be supplied is equal to the sum of the lost work. The make-up work to be supplied in operating a flow work exchanger is thus equal to the sum of lost work shown by the shaded areas in Fig. 3, lost work due to leakage of fluid due to volume inefficiencies of valves, and that due to the inefficiencies of the pumps $J_1$ and $J_2$.

The inefficiency of a flow work exchanger will be defined as the ratio of the make-up work, which must be supplied to the system, to the work exchanged between the two fluids. The inefficiencies of the flow-work exchangers will be described in connection with the actual performance of these units.

## V. THE FIRST FLOW-WORK EXCHANGER UNIT

### 1. Description of the Unit

Figures 4 and 5 show the front view and the side view respectively of the first flow-work exchanger unit. It is equipped with two floating-piston type displacement vessels actuated by a four-way hydraulic valve. The unit is operable up to 1500 psig and delivers 9 gpm. The component parts used are mostly products which are commercially available and have been adopted with some modifications. A list of the component parts used, their specifications and the names of the suppliers are summarized in Table 1. These component parts are connected by 3/4", schedule - 160 steel pipe in assembling the unit. Detailed descriptions of these component parts are given in Section VII.

Figure 6 gives a schematic illustration of the unit. This unit has not yet been connected to a reverse osmosis system. An appropriate location to make such a connection is shown by region R in the figure. The unit has been assembled essentially according to the scheme shown in Fig. 2. The following descriptions are given to supplement those given earlier in connection with Fig. 2.

a. Since the unit has not been connected to a reverse osmosis system, the volume of the high pressure fluid discharged from the system is essentially equal to the volume of the feed stream. Therefore, the amount of feed stream to be pumped in by the high pressure pump $J_3$ is very slight. However, a high pressure pump $J_3$ is still required in order to maintain the system at a desired pressure. By-pass line $B_1$ with relief valve $V_R$ is provided to discharge most of the fluid pumped by $J_3$ and return it to tank $T_2$. By properly

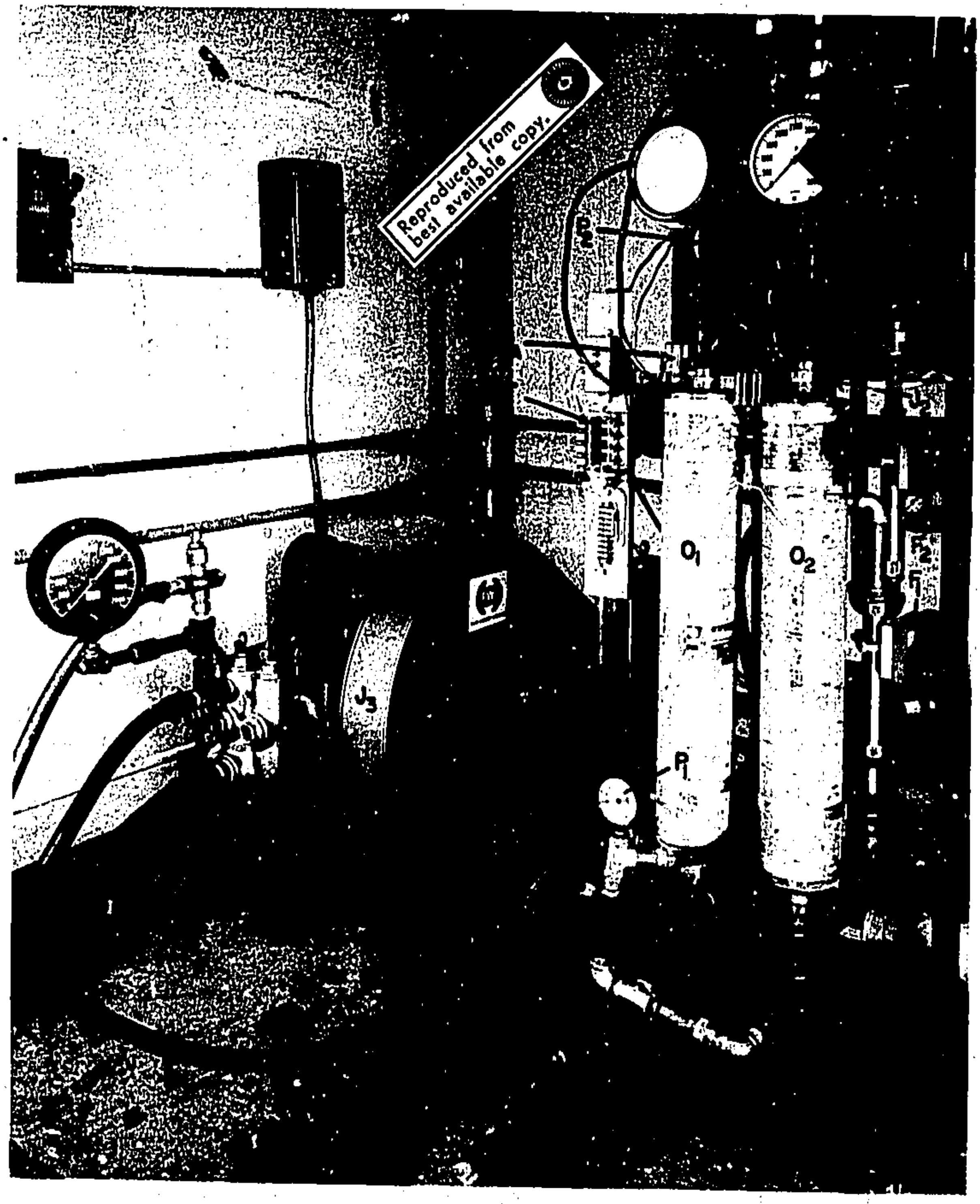



Fig. 4. Front view of the first flow-work exchanger unit.

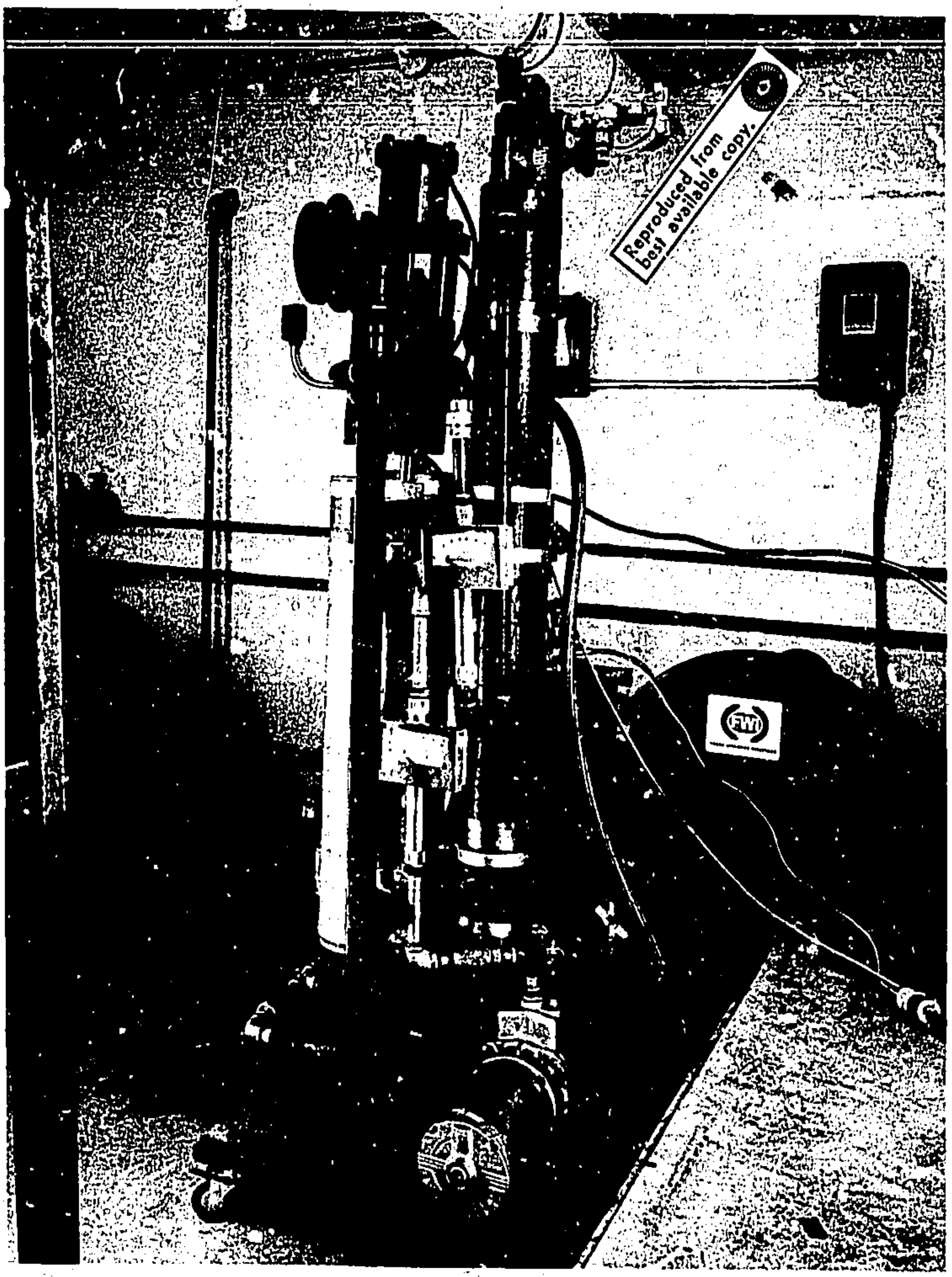



Fig. 5. Side view of the first flow-work exchanger unit.

TABLE 1. List of Component Parts used in the Flow-Work Exchangers Unit 1 and Unit 2

| COMPONENT PARTS | UNIT 1 | UNIT 2 |
|---|---|---|
| 1. Displacement Vessels, $O_1$ and $O_2$ | Floating piston type accumulator American Bosch Co., Springfield, Mass.<br>TYPE: ACC 2 gallons (Modified)<br>UNIT PRICE: $182.50 | Bladder type accumulator, Greer Olaer Products, Los Angeles, California<br>TYPE: A811-200 (Modified)<br>UNIT PRICE: $200.00 |
| 2. Control Valves, $V_5$, $V_6$, $V_7$ and $V_8$ | Hunt Hydraulic 4-way valve, Bellows-Valvair Co., Akron, Ohio<br>TYPE: 3/4" MS 551-P4, Series H 3000 psi<br>UNIT PRICE: $272.00 | Hunt hydraulic 4-way valve Bellows-Valvair Co., Akron, Ohio<br>TYPE: 1" MF 651-P4, Series G, 1500 psi<br>UNIT PRICE: $354.00 |
| 3. Check Valves, $V_1$, $V_2$, $V_3$, and $V_4$ | Hydraulic Check Valves*, Bellows-Valvair Co., Akron, Ohio<br>TYPE: A 401 05, 3000 psi<br>UNIT PRICE: $17.50<br>NO. REQUIRED: 4 | Hydraulic check valves*, Bellows-Valvair Co., Akron, Ohio<br>TYPE: A 401 06, 300 psi<br>UNIT PRICE: $17.50<br>NO. REQUIRED: 4 |
| 4. Low pressure, low head pump, $J_1$ | Burks pump<br>Decatur Pump Co., Decatur, Ill.<br>MODEL: 5HJS 1/2 HP<br>UNIT PRICE: $86.63 | Burks pump<br>Decatur Pump Co., Decatur, Ill.<br>MODEL: 10 W6<br>UNIT PRICE: $105.20 |

*These valves are for hydraulic oil service and are not suitable for water service.

| | | |
|---|---|---|
| 5a. High pressure low head pump, $J_2$ | Submergible pump, Reda Pump Co., Bartlesville, Okla. MODEL: 7D9P, 3/4 HP, 230 Volt, 1Ø, 3 Wire, with 460751 control box. UNIT PRICE: $275.31 | Submergible pump Reda Pump Co., Bartlesville, Okla. MODEL: 7D18P, 3/4 HP, 230 Volt 3Ø, with Allen-Bradley Starter UNIT PRICE: $220.00 |
| 5b. High pressure vessel to enclose $J_2$ | Accumulator without floating piston American Bosch Co., Springfield TYPE: ACC 3 gallons UNIT PRICE: $200.00 | Accumulator without floating piston American Bosch Co., Springfield TYPE: ACC 3 gallons UNIT PRICE: $200.00 |
| 6. Flow meters, $F_1$ and $F_2$ | Dial indicator purgemeter, Fischer & Porter Co., Warminister, Penn. MODEL: 10A 2227A, 1500 psi, 316S.S. 1" NPT, Max. flow 15 gpm. UNIT PRICE: $105.00 | Dial indicator purgemeter, Fischer & Porter Co., Warminister, Penn. MODEL: 10A 3565A, 1500 psi 1 1/2", Max. flo 50 gpm. UNIT PRICE: $170.00 |
| 7. Air control valve used in controlling valves $V_5$, $V_6$, $V_7$ and $V_8$ devoted as $V_A$ | Single solenoid pilot-operated valves, Speed King (No. 1432, 100 volt) Bellows-Valvair Co., Akron, Ohio UNIT PRICE: $52.00 | Single solenoid pilot-operated valves Asco Co., Florham Park, N. J. UNIT PRICE: $52.00 |
| 8. Programming timer, used to actuate air control valve, devoted as T. | Programming cam timers, Industrial Timer Corp., Los Angeles, Calif. MODEL: MC - 2 UNIT PRICE: $52.50 | Programming cam timers, Industrial Timer Corp., Los Angeles, Calif. MODEL: CM - 1 UNIT PRICE: $20.50 |
| 9. High pressure pump, $J_3$ | Triplex horizontal, single acting reciprocating plunger pump, aluminum bronze cylinder body, 1 1/8" plungers, 1780 psi. Frank Wheatley Industries, Tulsa, Oklahoma. MODEL: P-200A. UNIT PRICE: $1,667.16 | |
| 10. Relief valve, $V_R$ | Baird Co., Tulsa, Oklahoma, 1" Unit, UNIT PRICE: $70.00 | |
| 11. Pipe and pipe fittings | About $70.00 | |

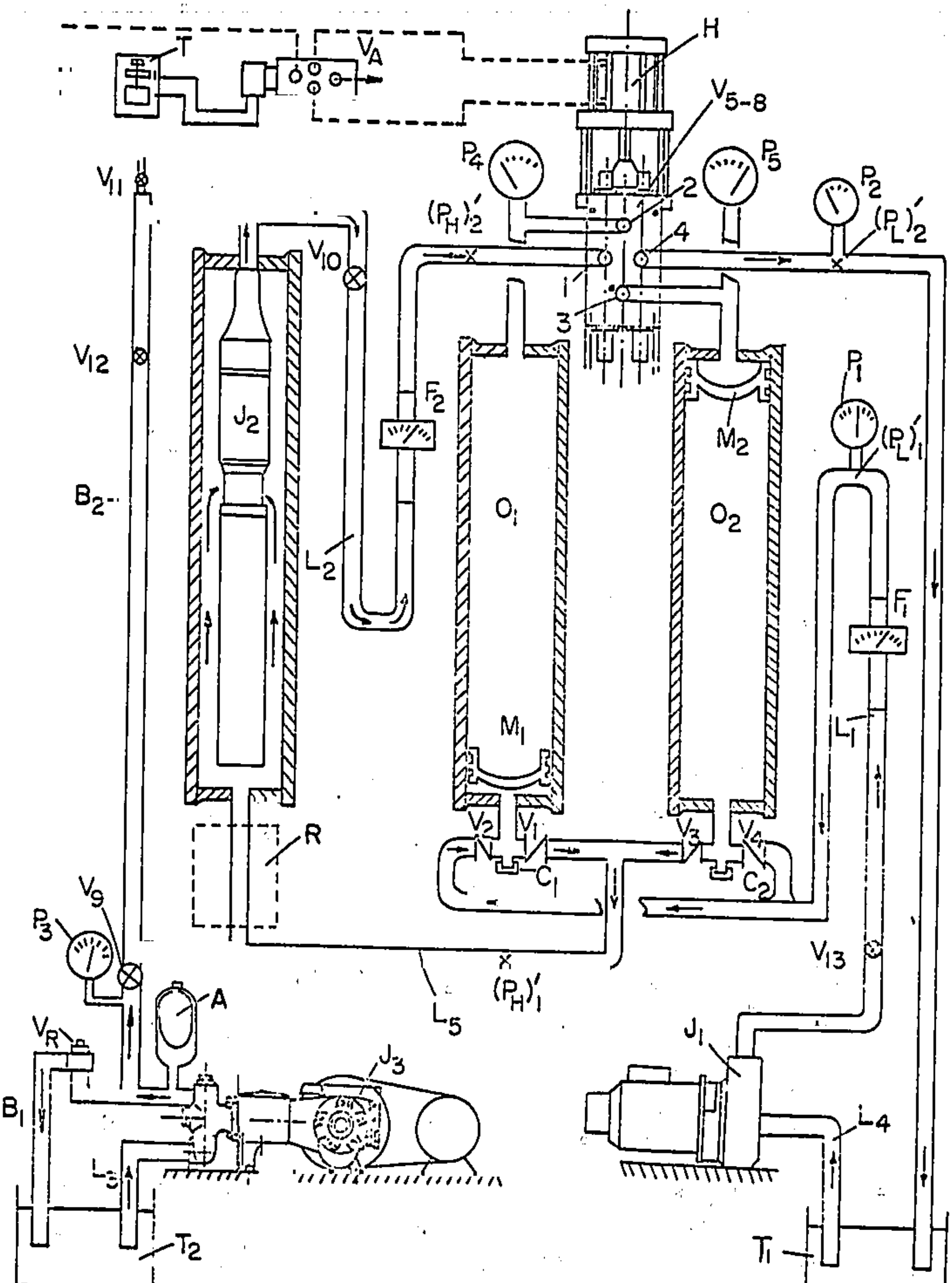



Fig. 6 . Schematic illustration of the first flow-work exchanger unit.

setting the relief valve $V_R$, the system pressure can be maintained at a desired value.

b. A four-way hydraulic valve, $V_{5-8}$, (see Fig. 6) is used in the unit in place of the four two-way valves, $V_5$, $V_6$, $V_7$ and $V_8$, shown in Fig. 2. This four-way valve is operated by a moving piston of a pilot cylinder H. The flow of compressed air (70-90 psi) is controlled by a solenoid-operated four-way air valve $V_A$ which again is operated by a programming timer T. The timer is adjusted to close the solenoid circuit during half of a cycle and to open the circuit during the remaining half of the cycle. The cycle time of the timer can be varied by changing the gear rack of the timer.

As shown in Fig. 6, ports 1 and 4 of the four way valve are connected to the high pressure line at pressure $(P_H)_2'$ and to the exhaust line leading to tank $T_1$ respectively, and ports 2 and 3 are connected to the displacement vessels $O_1$ and $O_2$ respectively. During half of a timer cycle, the piston of the pilot cylinder is at the upper position and during the remaining half, at the lower position. During the former half period, ports 1 and 4 are connected to ports 2 and 3 respectively, and, consequently, the displacement vessels $O_1$ and $O_2$ are at high pressure $(P_H)_2$ and low pressure $(P_L)_2$ respectively. Similarly, during the remaining half period, the displacement vessels $O_1$ and $O_2$ are respectively at low and high pressures.

c. In order to maintain continuous flow as far as possible, the movements of the floating pistons $M_1$ and $M_2$ in the displacement vessels $O_1$ and $O_2$ must be synchronized, and the flows in the lines $L_1$ and $L_2$ must be matched. Such a synchronizing control is not yet incorporated into this unit; rather, a manual control used during the test operation. Flowmeter $F_1$ is used to measure the flow in the low pressure line $L_1$ and flowmeter $F_2$ is used to measure the flow in the high pressure line $L_2$. In order to match the flows in the lines $L_1$ and $L_2$, branch line $B_2$ and two control valves, $V_{10}$ and $V_{12}$, are provided. A valve $V_{11}$ is provided for venting air from the system.

d. Submergible pump, $J_2$, is installed within a high pressure to serve as the high pressure circulation pump.

2. Operation of the First Unit

In operating the unit, we have to remove air from the system, adjust the system pressure to a desired operating condition and adjust the flow rates in lines $L_1$ and $L_2$ so as to synchronize the movements of floating pistons $M_1$ and $M_2$ and limit the flow rates so as not to exceed the capacity of the displacement vessels.

Start-up high pressure pump $J_3$.

First loosen relief valve $V_R$ and set it at a low pressure setting. Manually rotate the pump pulley to suck water through suction line $L_3$ and displace air from both the lines and the pump body. Power can now be supplied and the operating pressure can be adjusted by manupulating the valve setting of $V_R$.

Air venting from the unit

Most of the air in the system can be removed from air vent $V_{11}$ except the air trapped in the displacement vessels below the floating pistons. In a future unit we may provide a small hole in each floating piston or provide an air vent at the low end of each displacement vessel to facilitate air venting. In the present unit, air trapped in the displacement vessels below the floating pistons is removed as follows;

Referring to vessel $O_1$, we open cap $C_1$, pressurize the vessel to, say, 80 psi to force the piston down, insert a small tube until it touches the piston, and admit fluid to the lower end of the vessel by pump $J_1$. After the entrapped air is removed through the inserted tube, the opening is capped by $C_1$. A similar operation is applied to vessel $O_2$.

Adjustment of system pressure.

As has been described, system pressure can be maintained at a desired operating condition by adjusting the valve setting of the relief valve $V_R$.

Control of flow rates.

To maintain continuous flow within lines $L_1$ and $L_2$, except for the brief period of valve shifting, each of the flow rates multiplied by half of the timer cycle time should not exceed the volume swept by the piston in a stroke. Therefore, in order to increase flow rate, we have to adjust the programming timer and reduce its cycle time. Furthermore, the flow rates in lines $L_1$ and $L_2$ have to be adjusted to the same value. An automatic control device may be used, but in the present unit we simply control valves $V_{10}$, $V_{12}$ and $V_{13}$ manually.

3. Performance of the Unit

Testing has been done with water as the working medium. Water from tank $T_1$ enters pump $J_1$ through line $L_4$, passes through line $L_1$ at pressure $(P_L)_1'$, and enters displacement vessels $O_1$ and $O_2$, alternately, is pressurized and displaced by pressurized discharge water, enters a high pressure system R at pressure $(P_H)_1'$ (omitted in this unit), and enters recirculation pump $J_2$. Water is discharged from the pump $J_2$ at pressure $(P_H)_2'$ and enters displacement vessels $O_1$ and $O_2$ alternately through the control valve $V_{5-8}$. The water is thereby depressurized to pressure $(P_L)_2'$ and is discharged into tank $T_1$. Figure 7-A schematically shows the flow of water in the unit. It shows that the flow is steady except during the brief period of valve shifting. At 9 gpm, the cycle time J is adjusted to 20 seconds. The amount of fluid entering the vessels during a half cycle is 1.5 gallons, which is less than the maximum displacement of the piston per stroke, 1.8 gallons. By installing an accumulator along line $L_5$, the flow fluctuation can be smoothed out, and

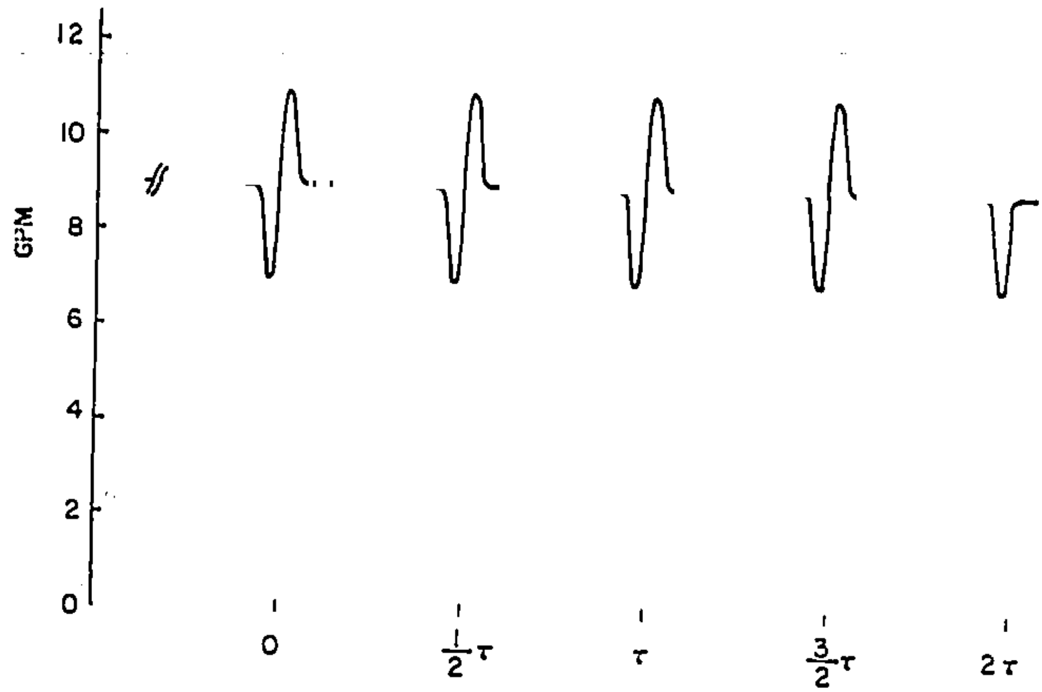



Fig. 7-a. Flow through the flow-meters in the first unit (schematic).

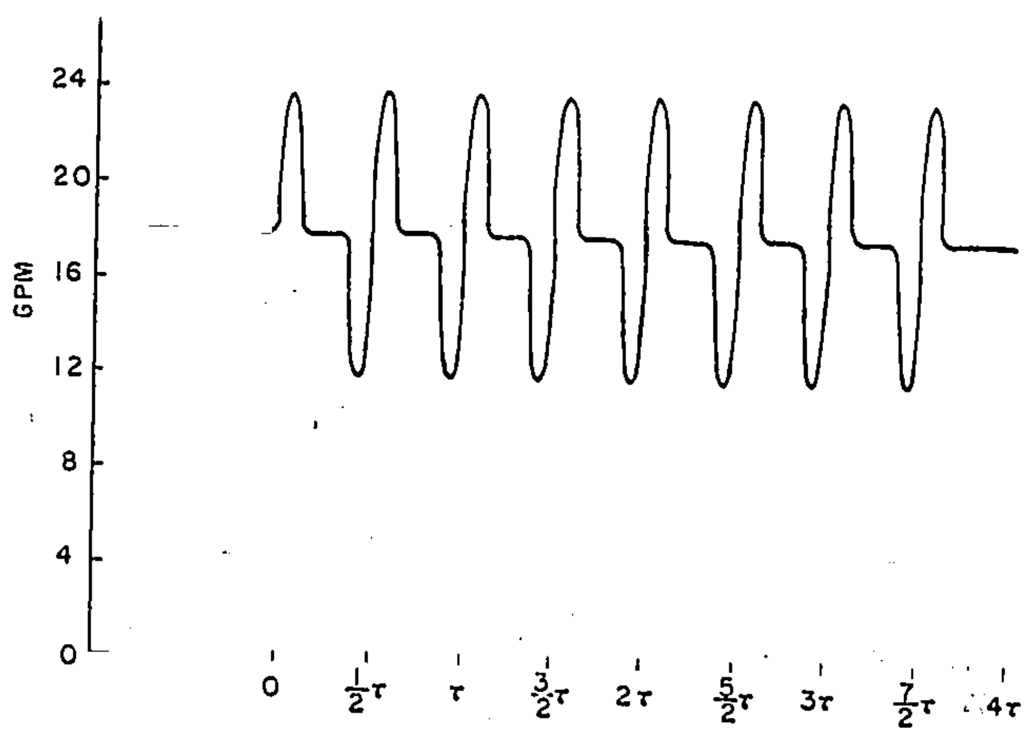



Fig. 7-b. Flow through the flow-meters in the second unit (schematic).

hydraulic shocking in the high pressure part of the system due to the quick opening and closing operations of valve $V_{5-8}$ can be removed.

The characteristic curve of the submergible pump (Reda Pump, 7D9P unit) is shown by line AB in Fig. 8. The curve drops very sharply at about 9 gallons per minute. Due to these operating characteristics, the delivery capacity of the first flow-work exchanger unit is limited to 9 gallons per minute. The operating characteristics of the control valve, $V_{5-8}$, have been studied by measuring the pressure drops, $(\Delta P)_1$, across pairs of ports 1-2, 1-3, 4-2 and 4-3, at various flow rates. The average value, $(\Delta P)_1$, is plotted against flow rate as line 1 in Fig. 9. The pressure drops across the check valves, $V_1$, $V_2$, $V_3$, and $V_4$, have also been measured at various flow rates and their average value, $(\Delta P)_2$, is also plotted against flow rate as line 2 in Fig. 9. The sum of these two pressure differentials, $(\Delta P)_1 + (\Delta P)_2$, is shown by line 3 in the figure.

The flow-work exchanger unit has been operated under various system pressures and at various flow rates. The pressure differential $(\Delta P)_{t,L} = (P_L)'_1 - (P_L)'_2$ as measured by the pressure gages $P_1$ and $P_2$ is recorded and tabulated in Table 1 and shown by line 4 in Fig. 9. The pressure drop required to overcome friction during this low pressure displacement operation $(\Delta P)_{D,L}$ is less than the $(\Delta P)_{t,L}$ value by an amount corresponding to the potential difference between the two pressure gages. Therefore,

$$(\Delta P)_{D,L} = (\Delta P)_{t,L} - \rho \Delta Z$$

$$= (\Delta P)_{t,L} - 1.2 \text{ psi.}$$

The $(\Delta P)_{D,L}$ values at various flow rates are also tabulated in Table 1 and shown by line 5 in Fig. 9. The $(\Delta P)_{D,L}$ value rather than the $(\Delta P)_{t,L}$ value should be used for $(P_L)_1 - (P_L)_2$ in Fig. 3 in evaluating the efficiency of the unit.

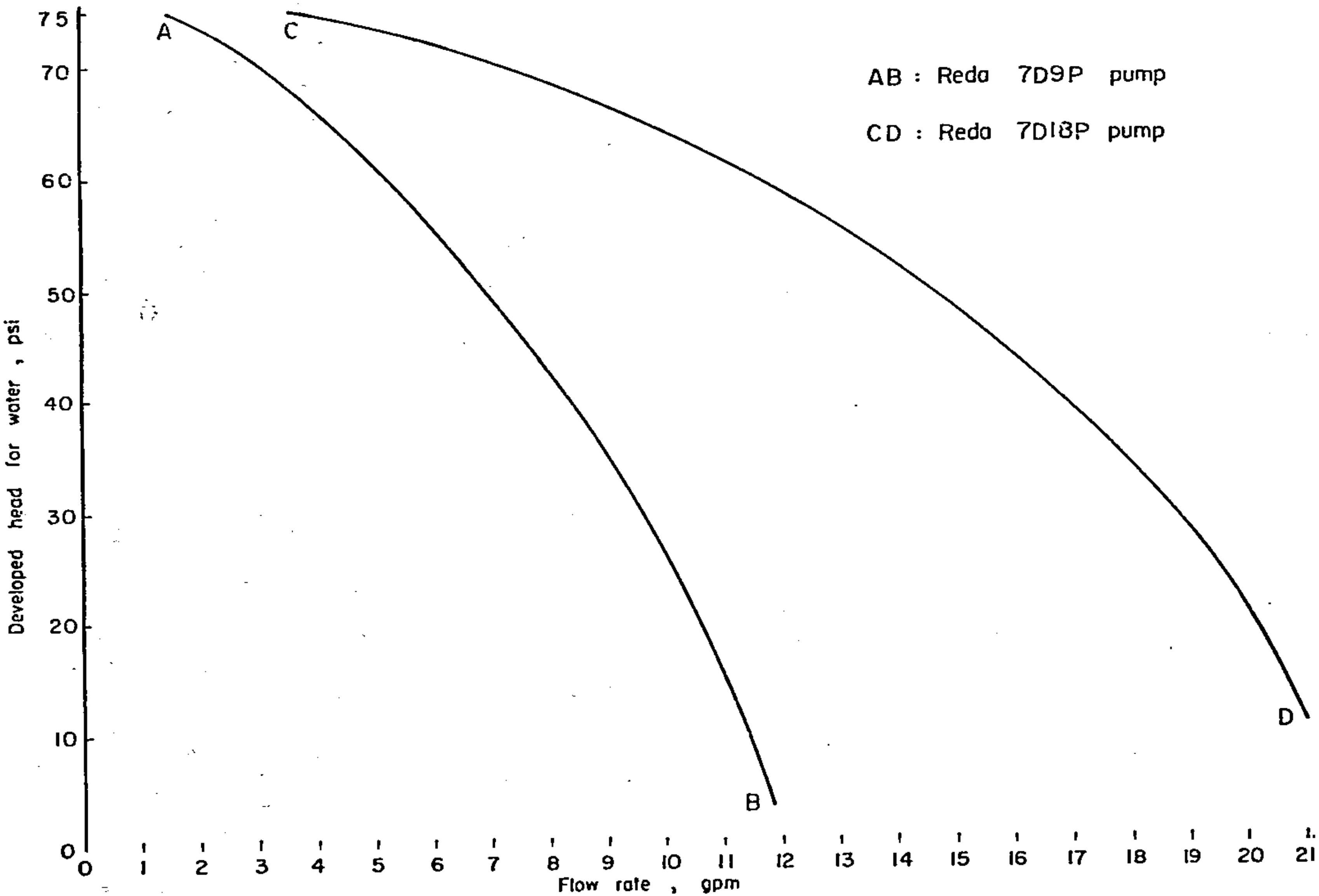



Fig. 8. The performance curves for the submergible pumps used in the first and second units.

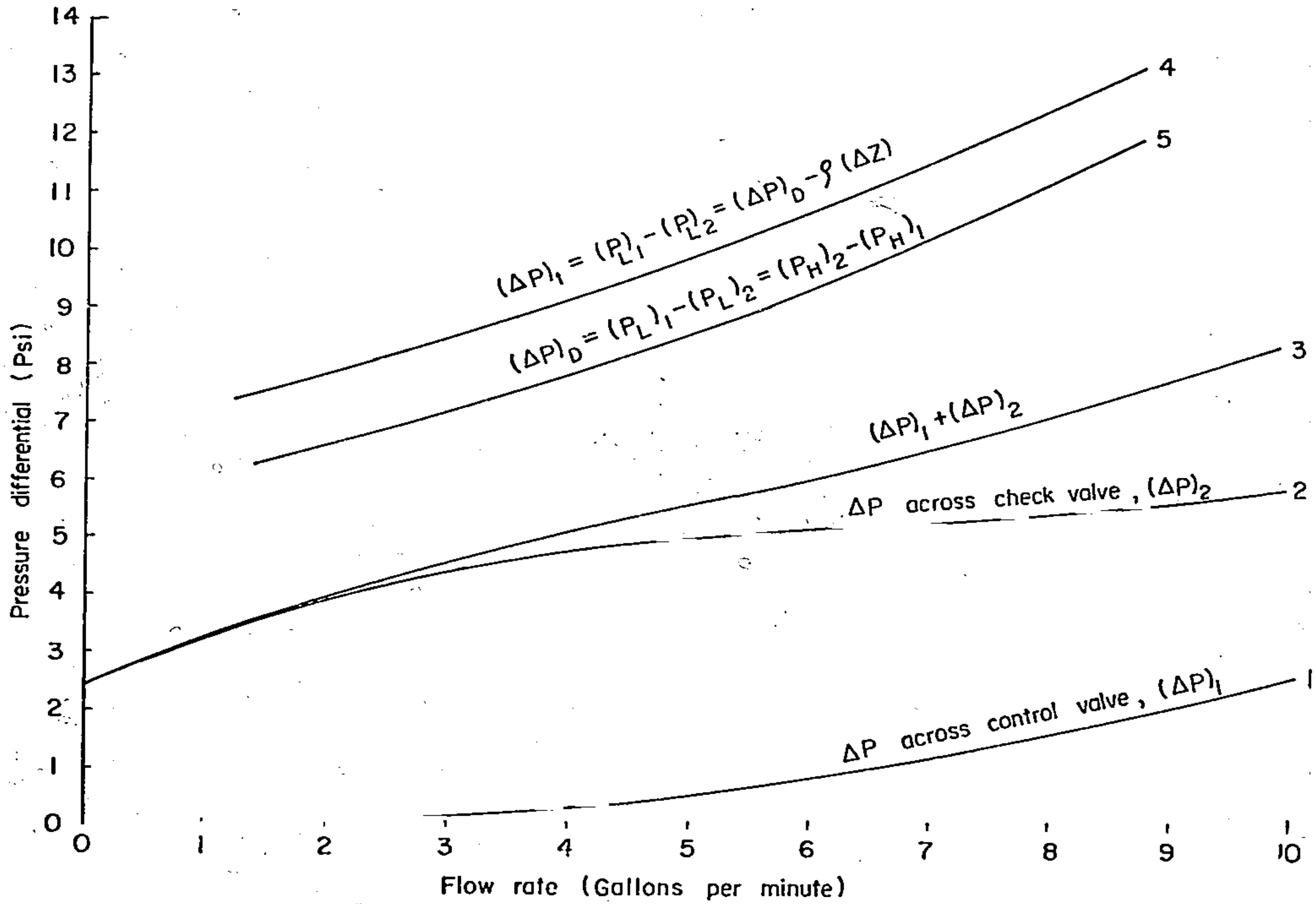



Fig. 9. Pressure drop vs. flow rate for the flow-work exchanger No.1 unit.

Table 2. Pressure differential required for displacement operations in Unit 1

Timer setting: 30 sec. per cycle

| System Pressure | Flow Rate | $(\Delta P)_t = P_1 - P_2$ | $(\Delta P)_D$ |
|---|---|---|---|
| 800 psi | 1.5 GPM | 7.5 psi | 6.3 |
| 800 | 3.0 | 8.0 | 6.8 |
| 800 | 6.0 | 10.0 | 8.8 |
| 800 | 7.5 | 12.0 | 10.8 |
| 800 | 9.0 | 14.0 | 12.8 |
| 1500 | 1.5 | 7.5 | 6.3 |
| 1500 | 3.0 | 8.5 | 7.3 |
| 1500 | 4.5 | 9.5 | 8.3 |
| 1500 | 6.0 | 11.0 | 9.8 |
| 1500 | 7.5 | 12.0 | 10.8 |
| 1500 | 8.5 | 13.0 | 11.8 |
| 1000 | 1.5 | 7.5 | 6.3 |
| 1000 | 3.0 | 8.5 | 7.3 |
| 1000 | 4.5 | 9.5 | 8.3 |
| 1000 | 6.0 | 11.0 | 9.8 |
| 1000 | 7.5 | 12.5 | 11.2 |
| 1000 | 8.5 | 14.0 | 12.8 |
| 1200 | 1.5 | 8.0 | 6.8 |
| 1200 | 3.0 | 8.5 | 7.3 |
| 1200 | 4.5 | 9.5 | 8.3 |
| 1200 | 6.0 | 10.5 | 9.3 |
| 1200 | 7.5 | 12.5 | 11.3 |
| 1200 | 8.5 | 13.5 | 12.3 |

The pressure drop required to overcome friction during a high pressure displacement operation, $(\Delta P)_{D,H}$, is essentially the same as the $(\Delta P)_{D,L}$ value, since during these two displacement operations fluid simply flows through the same system in the reverse direction.

It has been found by separate tests that, for the system of 1500 psi, the volume expansion (8-9 in Fig. 3) and the volume contraction (5-1 in Fig. 3) during non-flow depressurization and non-flow pressurization operations respectively are about 0.7% of the volume of the displacement vessel.

4. Efficiency of the Unit

When a pair of matching pumps are used for $J_1$ and $J_2$ and the unit is operated at such a capacity that the pumps are at their respective normal efficiencies and the valves $V_{10}$ and $V_{13}$ and the branch line $B_2$ are removed from the system, measurement of power inputs at pumps $J_1$ and $J_2$ will give us the make-up energy requirement. Since such is not the case in the operation of this unit, the make-up energy required and the efficiency of the unit have been evaluated by the procedure described in Section IV, allowing for the normal pump and motor efficiencies.

Referring to Fig. 3, the make-up energy required per cycle is given as the sum of areas 3-4-10-9, 4-5-1, 5-6-7-2, and 7-8-9. Referring to Fig. 9, the $(\Delta P)_D$ value at 9 gpm is 12.1 psi. Since the volume of each of the displacement vessels is 2 gallons, both areas 3-4-10-9 and 7-6-5-2 are

$$12.1 \text{ (psi)} \times 2 \text{ (gallons)} = 24.2 \text{ psi - gallons.}$$

As described, the volume expansion and contraction during non-flow pressurization and depressurization operations at the system pressure of 1500 psig are 0.7% of the volume of the displacement vessel. Therefore, both areas 4-5-1 and 7-8-9 are

$$1/2 \times 1500 \text{ (psi)} \times (2 \times 0.007) \text{ gallons} = 10.5 \text{ psi - gallons.}$$

Assuming that the pump efficiency is 70% and motor efficiency is 85%, the total make-up energy required is

$$(24.2 \times 2 + 10.5 \times 2) \times \frac{1}{0.7 \times 0.85} = 11.4 \text{ psi - gallons.}$$

Since the energy exchanged between the two fluids is

$$1500 \text{ (psi)} \times 2 \text{ (gallons)} = 3000 \text{ psi - gallons,}$$

the make-up energy required is

$$\frac{11.4}{3000} \times 100 = 3.8\%$$

of the energy exchanged. Considering that the volumetric efficiency of the values to be 98%, the make-up energy required will still be less than 6% of the energy exchanged.

Since most of the friction losses take place in the valves, pipe fittings and the connecting pipes, we can increase the capacity of the unit to about 15 gallons per minute at about the same efficiency by using 1" pipe, pipe fittings and valves in place of 3/4" pipe, pipe fittings and valves.

5. Cost of a Flow-Work Exchanger Unit

i. A 9 gpm unit

As described, the component parts used in building the first unit and their purchase prices (detailed price) are given in Table 1. Since we need two displacement vessesl, a control valve, 4 check valves, a low pressure circulation pump, a high pressure circulation pump and its high pressure enclosure, two flow meters, an air control valve, a programming timer and connecting pipes and pipe fittings, the total material cost is given by

182.50 x 2 + 272.00 + 17.50 x 4 + 86.63 + 275.31

+ 200.00 + 105.00 x 2 + 52 + 52.50 + 70 = $1653.44.

Assuming that it takes 5 man-days to assemble a unit and assuming a labor cost of $3.00 per hr., the total cost becomes

1653.44 + 3 x 8 x 5 = $1773.44.

In the above calculation, the cost of a high pressure pump $J_3$ is not included, because it is need anyway in pumping excess volume of feed (defined as volume of feed minus volume of high pressure discharge stream).

If flow meters are not included and a single cam programming timer (model CM-1) is used instead of the multiple cam timer (model MC-2), the cost will be reduced by the amount of

105 x 2 + 52.50 - 20.50 = $242

Then, the cost of a unit becomes

1773.44 - 242 = $1531.44.

Since most of these component parts can be purchased at 30% reduction bases for resale purposes, the cost calculated on these bases will be about $1110.

ii. A 20 gpm unit

As described, most of the friction drop takes place across the valves and pipe fittings, and the factor limiting the capacity of the unit is the capacity of the submergible pump. Therefore, by using a model 7D18P Reda pump and using 1" pipe, pipe fittings and valves, the capacity of the unit can be increased to about 20 gpm. The total material cost based on the purchase prices and including labor cost will be given by

182.50 x 2 + 354.00 + 17.50 x 4 + 105.20 + 220.00

+ 200 + 170 x 2 + 52 + 20.50 + 70 + 3 x 8 x 5 = $1916.70

Again if flow meters are not included, the cost becomes

$$1916.70 - 170 \times 2 = \$1576.70$$

If these component parts are obtained on resale-base, the cost becomes about $1140.00.

In comparing the cost of a flow-work exchanger with other power recovery means, it must be remembered that a flow-work exchanger unit functions as a combination of a pressurizer and a depressurizer.

## VI. THE SECOND FLOW-WORK EXCHANGER UNIT

### 1. Description of the Second Unit

Figures 10 and 11 respectively show the fron view and a side view of the second flow-work exchanger unit. It is equipped with two bladder-type displacement vessels actuated by a four-way hydraulic valve. The unit is operable up to 1500 psig and delivers 18 gpm. The component parts used are mostly commercially available products and have been adopted with some modifications. A list of the component parts used, their specifications, the names of the suppliers and the purchase prices are also summarized in Table 1. These component parts are connected by 1", schedule - 160 steel pipe in assembling the unit. Detailed descriptions of these component parts are given in Section VII.

Figure 12 gives a schematic illustration of the unit. This unit has not yet been connected to a reverse osmosis system. An appropriate location for such a connection is shown by the region R in the figure. The unit is also assembled essentially according to the scheme shown in Fig. 2. Therefore, this unit is similar to the first unit except that bladder-type displacement vessels are used in the place of floating piston type displacement vessels. The description made in connection with the first unit also applies here.

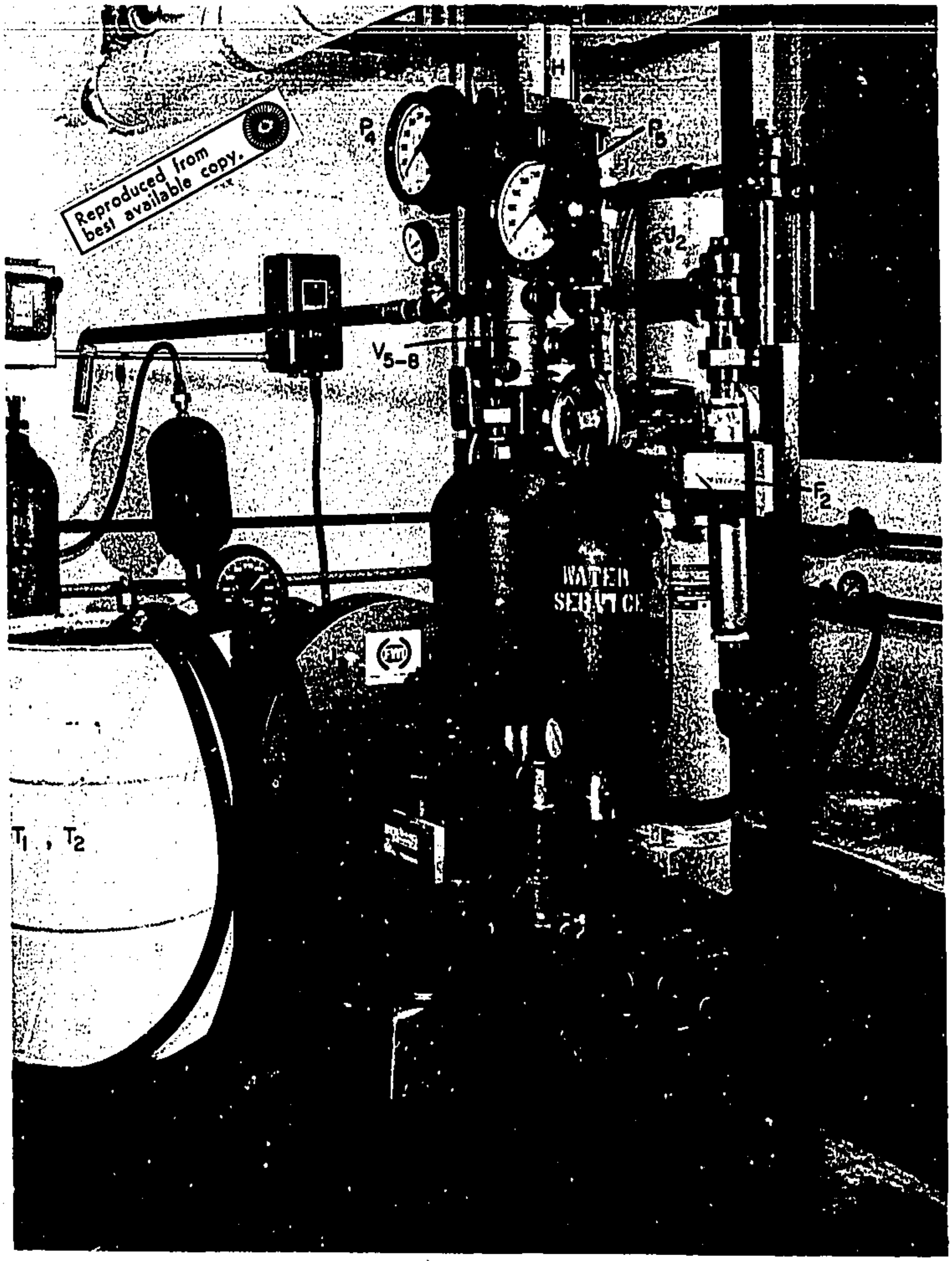



Fig. 10. Front view of the second flow-work exchanger unit.

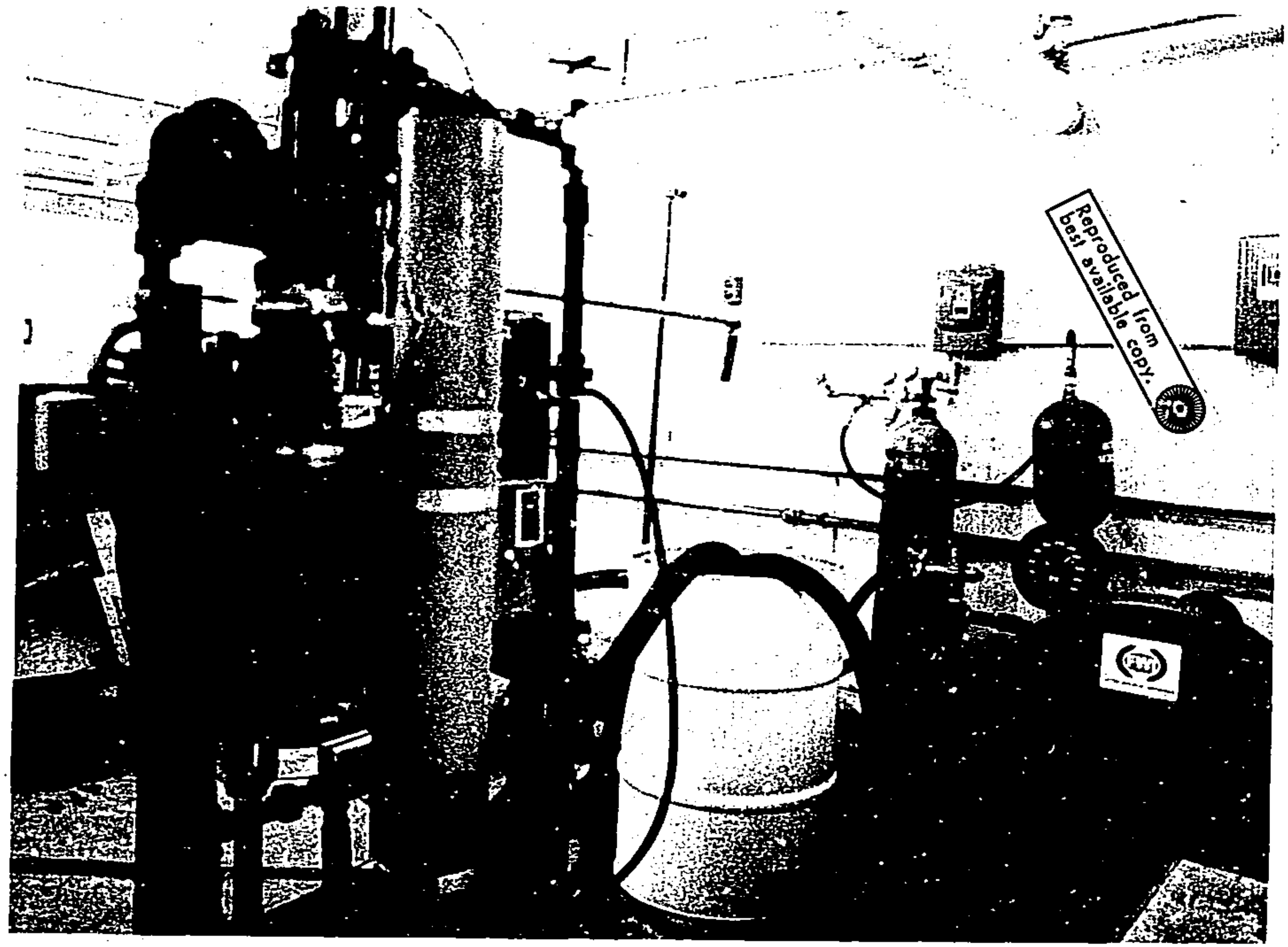




Fig. 11. Back view of the second flow-work exchanger unit.

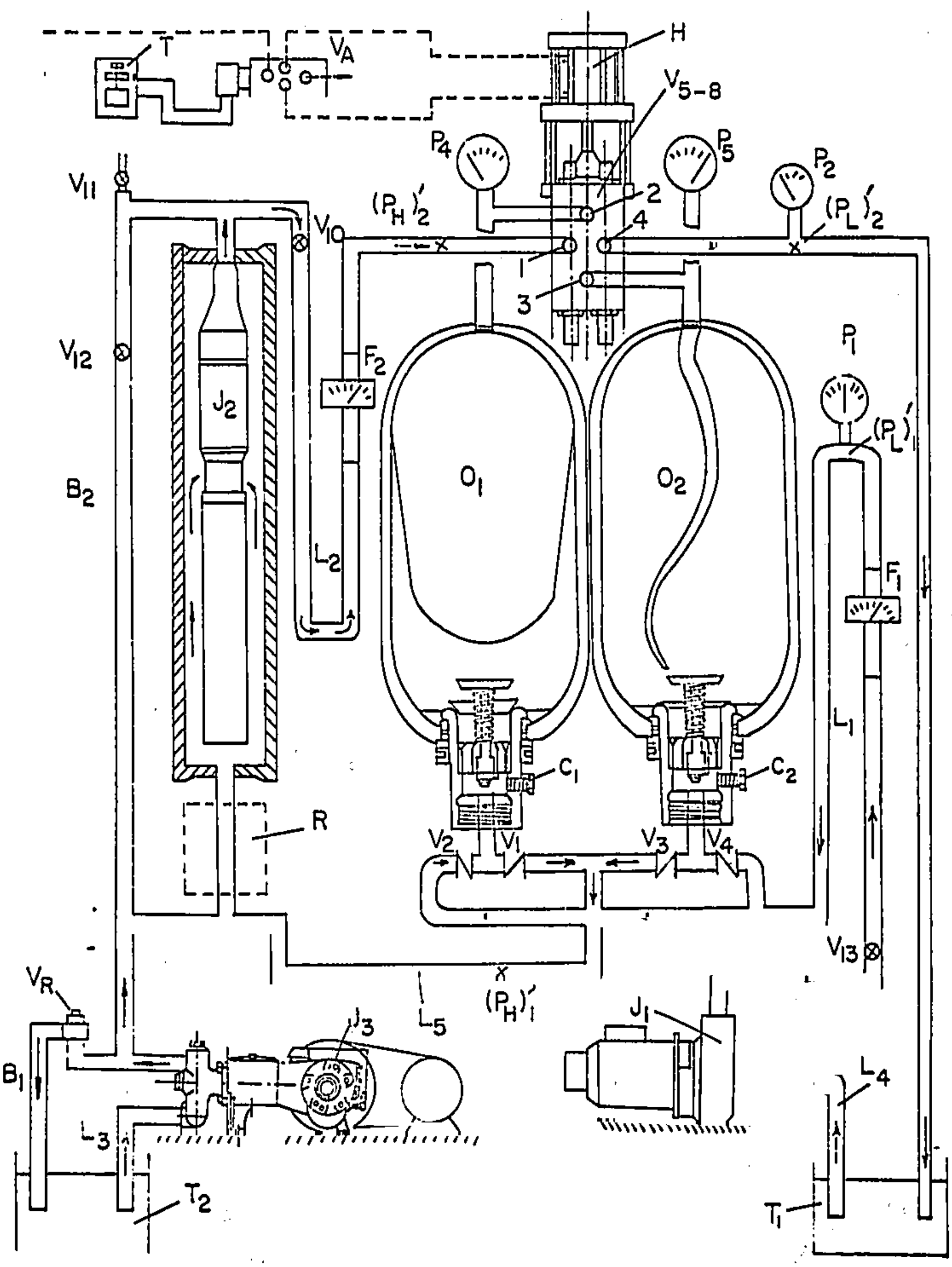



Fig. 12. Illustration of flow work exchanger unit No. 2.

2. Operation of the Second Unit

The descriptions relative to the first unit apply here by simply replacing the floating pistons $M_1$ and $M_2$ by the bladder $M_1$ and $M_2$. Therefore, the detailed descriptions are omitted. Air entrapped in the displacement vessels below the bladders are vented at the openings $C_1$ and $C_2$ shown in Fig. 12, which are then plugged off.

3. Performance of the Second Unit

Referring to Figure 7-B, the fluid flows in lines $L_1$ and $L_2$ are much less steady than in the case of the first unit. The flows measured by flow meters $F_1$ and $F_2$ shoot up at the beginning of each half cycle and tail off at the end of each half cycle. The unsteadiness in these flows are due to the following reasons: These displacement vessels have been obtained by modifying 2 1/2 gallon Greer accumulators. The rubber bags used in these accumulators are of only 1 1/2 gallon capacity. Therefore, the bladder in a displacement vessel stretches at the end of each high pressure displacement operation of the vessel and contracts at the beginning of each low pressure displacement operation of the vessel. These contracting and stretching actions of the rubber bladders cause the unsteadiness in the flows. This unsteadiness can be removed by installing bladders of capacity equal to the capacity of the outer shells. However, we were unable to obtain larger size bladders in time to improve the unit. It should be emphasized that the unsteadiness is not something inherent in the bladder type displacement vessels.

The characteristic curve of the submergible pump (Reda Pump, 7D9P unit) is shown by line CD in Fig. 8. The curve drops very sharply at about 20 gallons per minute. Due to this operating characteristic, the delivery capacity of the first flow-work exchanger unit is limited to 20 gallons per minute. The operating characteristics of the control valve, $V_{5-8}$, have been studied by measuring the pressure drops, $(\Delta P)_1$, across pairs of ports 1-2, 1-3, 4-2 and 4-3, at various flow rates. The average value, $(\Delta P)_1$, is plotted against flow rate as line 1 in Fig. 13. The pressure drops across the check valves, $V_1$, $V_2$, $V_3$, and $V_4$, have also been measured at various flow rates and their average value, $(\Delta P)_2$ is plotted against flow rate as line 2 in Fig. 13. The sum of these two pressure differentials, $(\Delta P)_1 + (\Delta P)_2$ is shown by line 3 in the figure.

The flow-work exchanger unit has been operated under 1500 psig and at various flow rates. The pressure differential $(\Delta P)_{t,L} = (P_L)_1' - (P_L)_2'$ as measured by the pressure gages $P_1$ and $P_2$ is shown by line 4 in Fig. 13. The pressure drop required to overcome friction during this low pressure displacement operation $(\Delta P)_{D,L}$ is less than the $(\Delta P)_{t,L}$ value by an amount corresponding to the potential difference between the two pressure gages. Therefore,

$$(\Delta P)_{D,L} = (\Delta P)_{t,L} - \rho\Delta Z$$

$$= (\Delta P)_{t,L} - 1.2 \text{ psi.}$$

The $(\Delta P)_{D,L}$ values at various flow rates are shown by line 5 in Fig. 13. The $(\Delta P)_{D,L}$ value rather than the $(\Delta P)_{t,L}$ value should be used for $(P_L)_1 - (P_L)_2$ in Fig. 2 in evaluating the efficiency of the unit.

As explained earlier, the pressure drop required to overcome friction during a high pressure displacement operation, $(\Delta P)_{D,H}$, is essentially the same as the $(\Delta P)_{D,L}$ value.

4. Efficiency of the Second Unit

When a separating bag of a size equal to or slightly larger than the outer shell is used, there will be no stretching of the bag, and the liquid

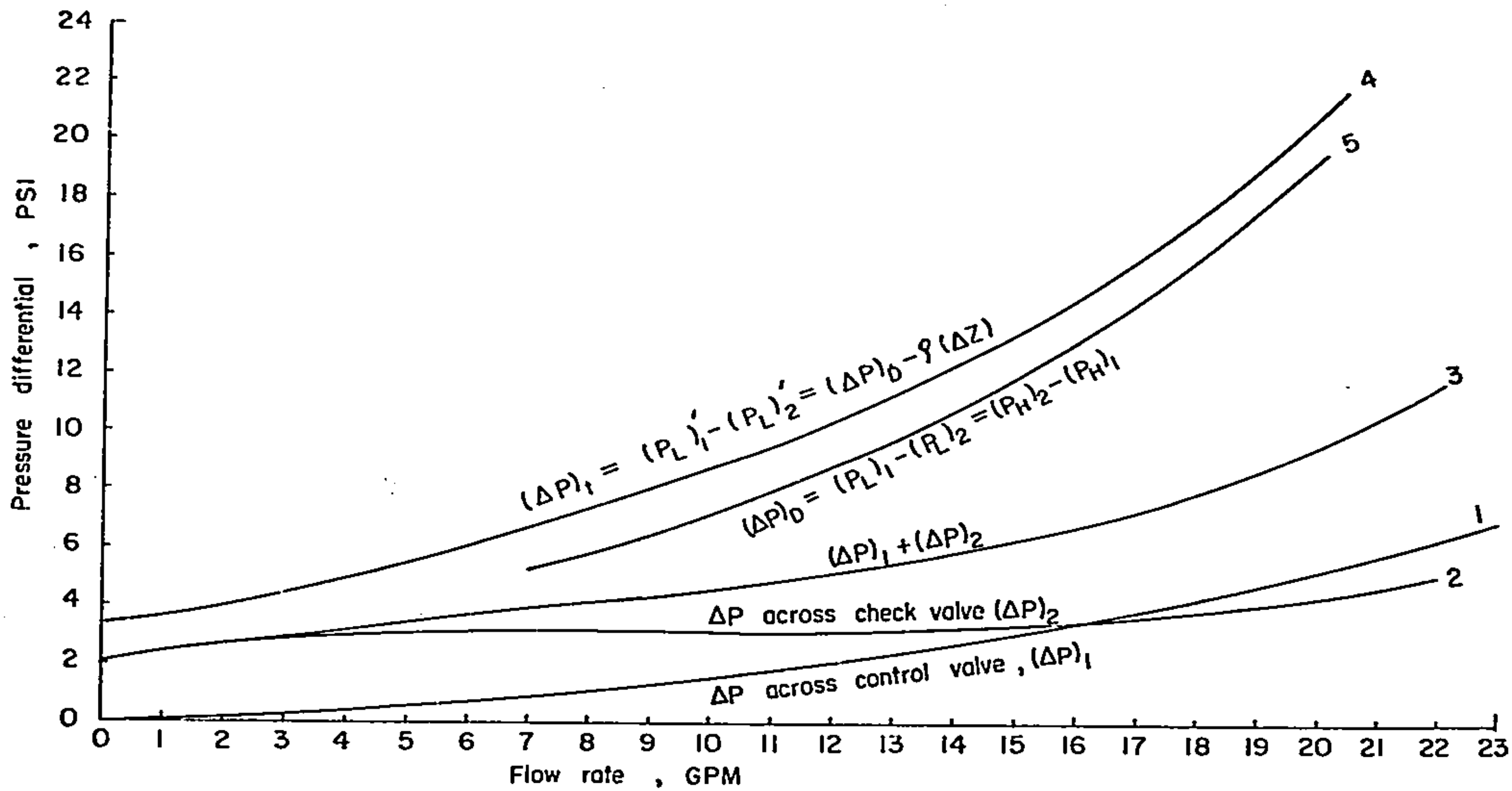



Fig.13 Pressure drop vs. flow rate for the flow-work exchanger No. 2 unit.

flow will be as smooth as that of a floating-piston type unit. Since the pressure differential required for the bag movement is very low, it is expected that the efficiency of a bladder type unit will be slightly better than that of a floating piston type. The make-up energy required is on the order of 6% of the energy exchanged at 1500 psi and 20 gpm.

5. Cost of a Flow Work Exchanger Unit

The cost of a bladder type flow-work exchanger unit is about the same as that of a floating piston type of equivalent capacity.

## VII. DESCRIPTION OF PROMISING COMMERCIAL PRODUCTS WHICH CAN BE ADOPTED AS THE COMPONENT PARTS OF A FLOW-WORK EXCHANGER UNIT

A rather extensive search has been made for those commercial products which can be used as the component parts, such as the displacement vessels, high pressure circulation pump, control valves, check valves and timer. Various alternatives have been found usable. The descriptions of these products and the modifications required of these products in adopting them are summarized in this section.

1. Displacement Vessels

Displacement vessels can be classified into three types, viz. a floating piston type, b. bladder-type and c. diaphragm type. These vessels can be obtained by minor modification of hydro-pneumatic accumulators. Edward M. Greer (13), President of Greer Hydraulic Inc. has published a "Guide to Hydro-pneumatic Accumulators," which is a very good reference.

1a. Floating piston type vessels

Figure 14-a shows a hydraulic accumulator which is designed mostly for hydraulic oil service. It has a liquid discharge port and a gas valve. The pressure differential required to move the floating piston is about 15 psig. The modifications required for use as a displacement vessel are:

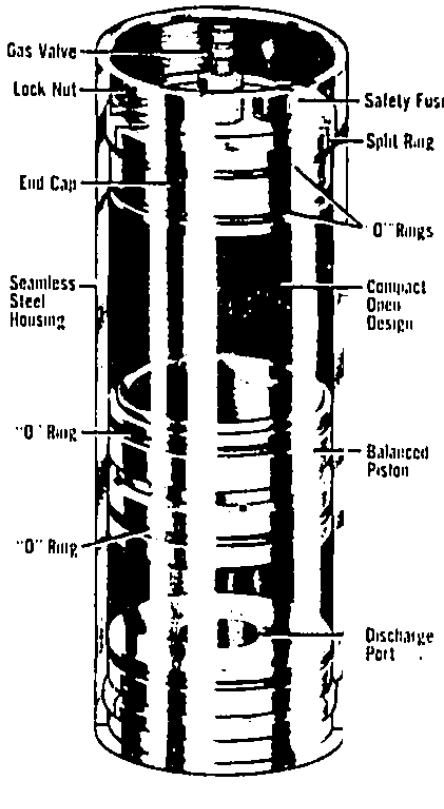



Fig. 14A. A piston type hydraulic accumulator.

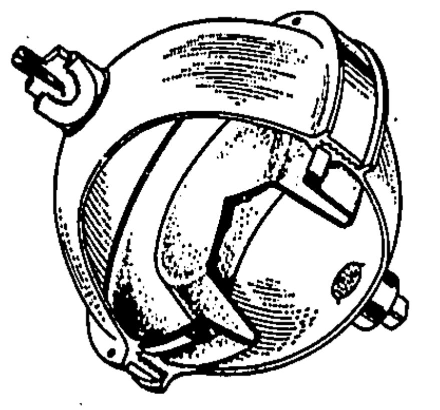


Fig. 14C. A diaphragm type aircraft accumulator.

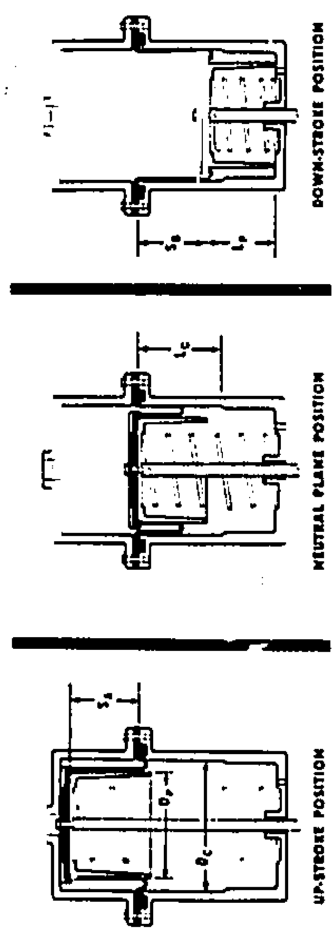



Fig. 14D. A rolling-type diaphragm air cylinder.

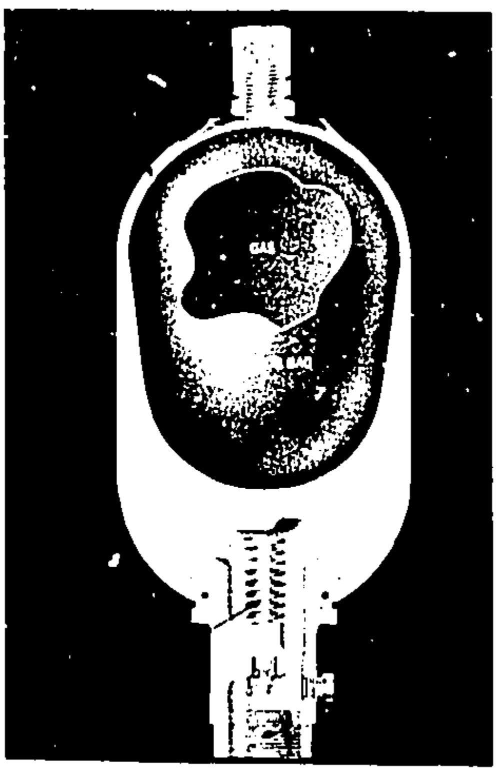



Fig. 14B. A bladder type hydraulic accumulator.

i. The inside surface should be nickel-plated or otherwise made corrosion resistant.

ii. The gas valve should be replaced by another liquid discharge port so that a pipe connection can be made.

iii. The pressure differential required for the floating piston should be reduced from 15 psi to about 2 psi. Referring to the figure, the floating balanced piston has two grooves with O-rings installed in the grooves. By enlarging the grooves slightly, the piston movement can be loosened. Another alternative is to replace the original piston by a water pump piston with leather caps.

The suppliers of piston type hydraulic accumulators are:

American Bosch Arma Bosch, Springfield, Mass.

Parker Hannifin, Des Plains, Ill.

Liquidonics, Inc., L. I., N. Y.

Mar-Oil Hydraulics, Hoboken, N. J.

Greer Hydraulics, Inc., Los Angeles, Calif.

1b. Bladder type vessels

Fig. 14-b, shows a bladder type hydraulic accumulator. It also has a gas valve and a liquid port. The holding capacity of the separating bag under unstretched conditions is smaller than the volume of the outer shell. For example, for a 2.5 gallon accumulator, the bag size is in the range of 1 - 1.5 gallons. Therefore, when the pressure inside the bag is larger than the pressure outside, the bag stretches. The poppet valve installed in the accumulator is a very good safety device in preventing the bag from bursting. The modifications required for use as a displacement vessel are:

i. The inside surface should be either plastic coated or plated to prevent corrosion.

ii. The gas valve should be replaced by a liquid port so that a pipe connection can be made. A specially made separating bag with a nipple of the desired size embedded in the rubber of the separating bag should be used.

iii. The separating bag should be of a size equal to that of the outer shell. This prevents the stretching of the separating bag and insures smoother liquid flow.

The suppliers of bladder type hydraulic accumulators are:

Greer Hydraulics, Inc., Los Angeles, Calif.

Vickers, Troy, Michigan.

1c. Diaphragm type vessels

Figure 14-c shows a diaphragm type air-craft accumulator. The modifications required for its use as a displacement vessel in a reverse osmosis system are

i. Coat the inner surface to prevent corrosion

ii. Replace the gas valve by a fluid port.

This type of accumulator has been described by E. M. Greer in the publication mentioned earlier and may be available from Greer Co. (13). An accumulator of this type should work satisfactorily.

The Warren Rupp Company (12) has recently started production of the so-called "Dynaflex Pump," which uses diaphragm type displacement vessels. The vessels are made only to withstand ordinary pumping pressure.

Rolling-type diaphragms are of relatively recent development (14). Bellofram Corporation, Burlington, Mass. manufactures such rolling-type diaphragms under the tradename "Bellofram." Rolling-type diaphragms have an extremely thin sidewall made of an elastomer and a fabric, each separately formed and then brought together in a mold to form a unique membrane. The wall usually has a thickness ranging from 0.008" to 0.055". Figure 14-d illustrates the operation of an air cylinder equipped with a rolling type diaphragm. It is seen that the wall of the diaphragm is supported either by the inside wall of the cylinder or the outside wall of the piston except at the rolling point. The height of the piston is about 1/3 of the cylinder length and the stroke is 2/3 of the cylinder length. By removing the piston shaft and installing fluid ports at the two ends, this type of vessel can also be used as a displacement vessel in a flow-work exchanger unit. However, since the stroke is 2/3 of the cylinder length, 1/3 of the cylinder volume is not effectively used.

2. High Pressure Circulation Pumps

Three types of pumps have been found usable as a high pressure circulation pump. They are a canned pump, a magnetic drive centrifugal pump, and a submergible pump. The canned pump is very expensive -- Integral Motor Pump Corporation has estimated a 10 Gpm pump for sea water service operable under 2000 psi at $4,500.00. This extremely high cost has prevented us from adopting it in the units built. The submergible pump is a low cost pump. The chief disadvantage of a submergible pump is that it has to be installed within a high pressure vessel and the vessel adds extra weight to the unit. The magnetic drive pump is reasonably low in cost and convenient to use. The main disadvantage of the magnetic drive pump is its low efficiency. The efficiency of a magnetic drive pump at its optimum operating conditions is about 50%. These pumps are briefly described as follows:

2a. Canned pumps

Referring to Fig. 15-a, it is seen that a canned pump consists of a motor and a pump, within a single leak-tight container (15). It differs from a mechanical pump mainly in that there is no packing gland -- one of the most troublesome parts of a conventional pump.

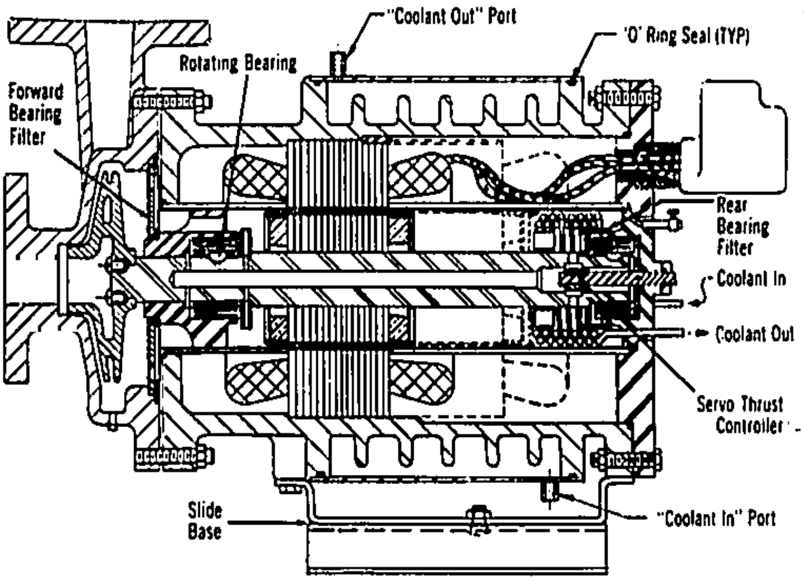



Fig. 15A. A zero leakage canned pump.

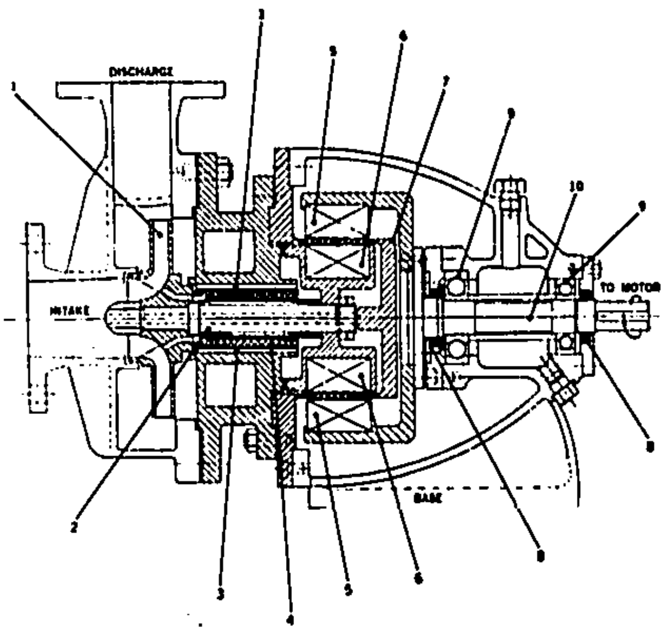



Fig. 15B. A Seal/Less magnetic drive pump.

Fig. 15C. A submergible pump.

Elimination of the packing gland is achieved by providing two "cans." The stator can consists of a stainless steel (or other alloy) tube, welded to a flange at one end and to a closure disc of solid stainless at the other. The flange of the stator can be sealed at the "wet end," or volute, of the pump, generally by means of Teflon gasket. The rotor of the motor is also "canned" by surrounding it with a welded stainless (or other alloy) jacket to prevent corrosion.

The rotor is supported on a shaft which rotates on bearings, which are graphite for such fluids as water or Freon, but may be filled Teflon for extreme oxidation resistance, or roller bearings for lubricating fluids. Graphite and Teflon bearings are cooled and lubricated by passing a portion of the fluid being pumped through slots in the bearings. In one variation called a "captive fluid pump," the bearings are cooled by providing an auxiliary impeller within the motor housing, with only a small pressure equalizing port into the motor. This auxiliary impeller circulates fluid in closed circuit within the motor housing to obtain bearing cooling and lubrication.

These pumps provide advantages other than zero leakage, with its attendant freedom from corrosion and packing maintenance problems. They are very much more compact than conventional pumps, permitting pipe mounting or smaller sizes, no foundations are needed for pumps as large as 30 horsepower.

2b. Magnetic drive pumps

F. Klans, Bocham, W. Germany manufactures a magnetic drive centrifugal pump under the tradename of "Seal/Less pump." Its distributor, the Kontro Company, Inc. (16) indicates that the pump is comparatively new in this country with three installations having been made by the company during the past year. Two of these installations were in pressurized systems at 1500 psi, and one at 300 psi. These were for high pressure, high temperature water service in connection with nuclear reactor test facilities and with special high pressure molding equipment. It has been reported that high pressure circulation operation is the field in which this pump really shines.

Referring to Fig. 15-b, a magnetic drive pump has drive magnets (5 in the figure) and driven pump magnets (6 in the figure), separated by a pump liquid containment shell made of thin non-magnetic material. Both the drive magnets and the driven pump magnets are powerful, permanent magnets. For very high pressure this shell is made of a super strength "NIMONIC" alloy which also has very good corrosion resistance properties.

A magnetic drive pump with casting and impeller constructed of #316 stainless steel delivering 40 GPM at 48 ft head operable under 1500 psig has been quoted at $3203. As has been described, the efficiency of a magnetic drive pump at its optimum operating condition is only about 50%.

2c. Submerged pumps

Referring to Fig. 15-c. a submergible pump consists of an oil filled electric motor, a multistage diffuser type pump, an oil filled capacitor and an oil reservoir and pressure equalizer. It has been recommended by Reda Co., that for a high pressure application the oil filled capacitor should be removed from the unit and placed outside of the high pressure system. The oil reservoir serves the purpose of equalizing the pressure. Therefore, the submergible pump can be installed within any high pressure system. Manufacturers of submergible pumps are

Reda Pump Co., Bartlesville, Oklahoma

Dayton Electric Mfg. Co., Chicago, Ill.

3. Directional Control Valves, $V_5$, $V_6$, $V_7$ and $V_8$

It has been mentioned in connection with Fig. 2 that four two-way directional control valves are needed in operating a flow-work exchanger unit. These valves may be replaced either by two three-way valves or one four-way valve. It is probably best to use a four-way valve, because of its simplicity in operation and low cost. It has been found that pilot operated double plunger valves work well. These valves are available on the market up to 6" port size. It has also been found that three-way ball valves and three-way poppet valves can be used. Other directional control valves, such as spool-type valves and shear-type valves, which are commonly used in oil hydraulics are not suitable for water service because of the low lubricity of water. Rather detailed descriptions of directional control valves available on the market are given in the "Fluid Power Handbook and Directory," 1968-1969 (17). The promising directional control valves are described as follows:

3a. Pilot operated double-plunger four-way valves

Figure 16A shows a pilot operated Hunt double plunger hydraulic valve and Fig. 16B shows a sectional view of the valve (18). Hunt double plunger valves are designed for use in water, water with soluble oil, or oil service. Three pressure ranges are available. Valves with the prefix letters MF, MSA and HH are rated respectively at 0-2000 psi, 0-3000 psi and 0-5000 psi. MF valves have bronze housings and both the MSA and HH valves have steel block housings. For the reverse osmosis sea water desalination, MF valves are recommended. Referring to the figures, the valve is actuated by an air cylinder. The plungers are in completely hydrostatic balance and -- since the pressure applied to the open side of the "U" packing is equal to the pressure coming from the radial ports of the plunger -- the packings are floated while

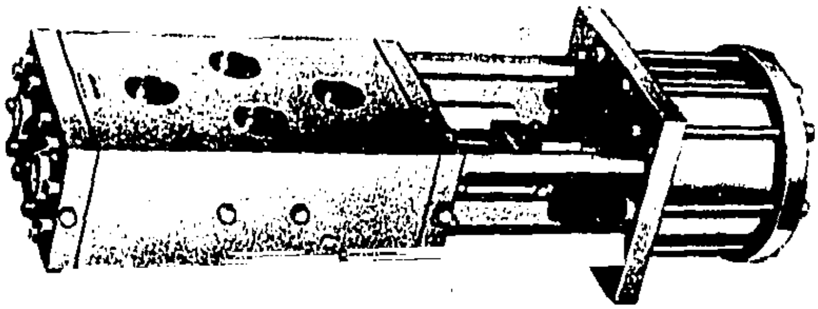


Fig. 16A. A pilot operated Hunt double plunger hydraulic valve.

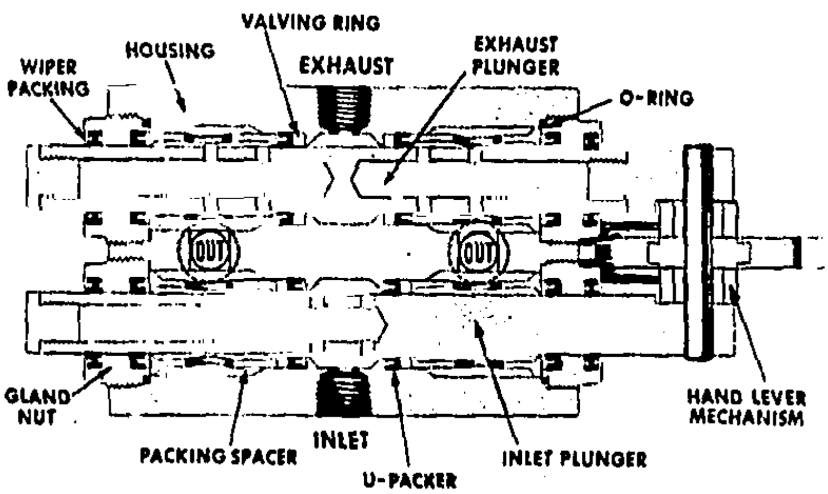



Fig. 16B. Sectional view of Hunt double plunger hydraulic valve showing the principal parts.

Fig. 16C. A 3-way ball valve.

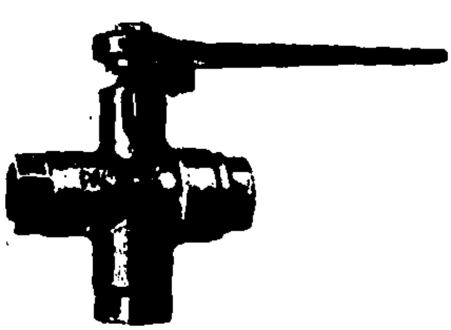
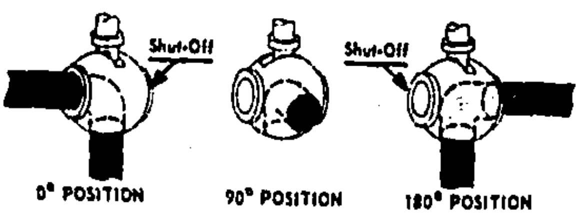


Fig. 16D. 3-way flow pattern for a 3-way ball valve.



crossing the valve ports. These valves are not designed for throttling. Two-position valves as used in a flow-work exchanger unit should always be shifted to their extreme positions.

Referring to Figures 6 and 12, ports 1 and 4 of the four way valve of a flow-work exchanger unit are connected to the high pressure line and the exhaust line respectively and ports 2 and 3 are connected to the displacement vessels $0_1$ and $0_2$ respectively. The first plunger, the exhaust plunger, has two hollow compartments, A and B, which have radial ports $a_1$ and $a_2$ respectively, and $b_1$ and $b_2$. The second plunger, the inlet plunger, has a hollow compartment C which has radial ports $c_1$ and $c_2$. When the plungers are shifted to the left extreme end, radial ports $c_2$ and $b_1$ are respectively connected to ports 1 and 4 and the radial ports $c_1$ and $b_2$ are respectively connected to ports 2 and 3. Therefore, high pressure fluid enters displacement vessel $0_1$ by passing through port 1, radial port $c_2$, hollow compartment C, radial port $c_1$, and port 2 and into the vessel $0_1$, and the fluid in the displacement vessel $0_2$ is discharged by passing through port 3, radial port $b_2$, hollow compartment B, radial port $b_1$ and port 4. When the plungers are shifted to the right extreme end, radial ports $c_1$ and $a_2$ respectively are connected to ports 1 and 4 and the radial ports $a_1$ and $c_2$ are respectively connected to ports 2 and 3. Therefore, high pressure fluid enters displacement vessel $0_2$ by passing through port 1, $c_1$, C, $c_2$ and port 3, and the fluid in the displacement vessel $0_1$ is exhausted by passing through port 2, $a_1$, $a_2$, and port 4. Hunt valves are available up to a 6" port.

The company has also introduced Hunt V-33 valves. These valves combine the proven features of the original Hunt hydraulic valves with a new design and a cartridge-type construction. These valves are currently more expensive than the original Hunt valves.

3b. 3-way ball valves

Figure 16C shows a three way ball valve and Fig. 16-D shows the 3-way flow pattern for a 3-way vall valve. Each displacement vessel requires a 3-way valve to control fluid flow. Port 1 of the valve is connected to a displacement vessel and ports 2 and 3 are respectively connected to a high pressure line and an exhaust line. When the ball is at the 0° position and 180° position, the displacement vessel is respectively connected to the high pressure line and the exhaust line. At the 90° position, the displacement vessel is shut off both from the high pressure and the exhaust lines. A manufacturer of 3-way ball valves is

Pacific Valves, Inc. Long Beach, Calif.

The operations of the two valves installed on two displacement vessels of a flow-work exchanger unit have to be synchronized.

Figure 17A shows a "Rota-Cyl" rotary actuator manufactured by Graham Engineering (19), and Fig. 17B shows a Flo-Tork rotary actuator manufactured by Flo-Tork, Inc. (20). These valves are powered by high pressure fluid and may be used in actuating the three way ball valves described above.

3c. 3-way poppet valves

Two 3-way poppet valves, Sinclair-Collins Valves of Bellows-Valvair Co., have been used in constructing the preliminary test unit described earlier. These valves have functioned properly. The disadvantages of these valves are i. the pressure drop is too high and ii. they are available only up to 2" port size. It is believed that 4-way poppet valve of larger sizes and lower pressure drop can be manufactured and used in the construction of flow-work exchanger units.

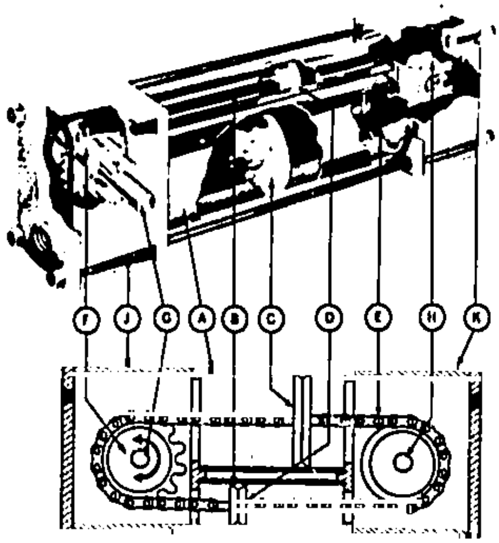



Fig. 17A. A ROTA-CYL rotary actuator.

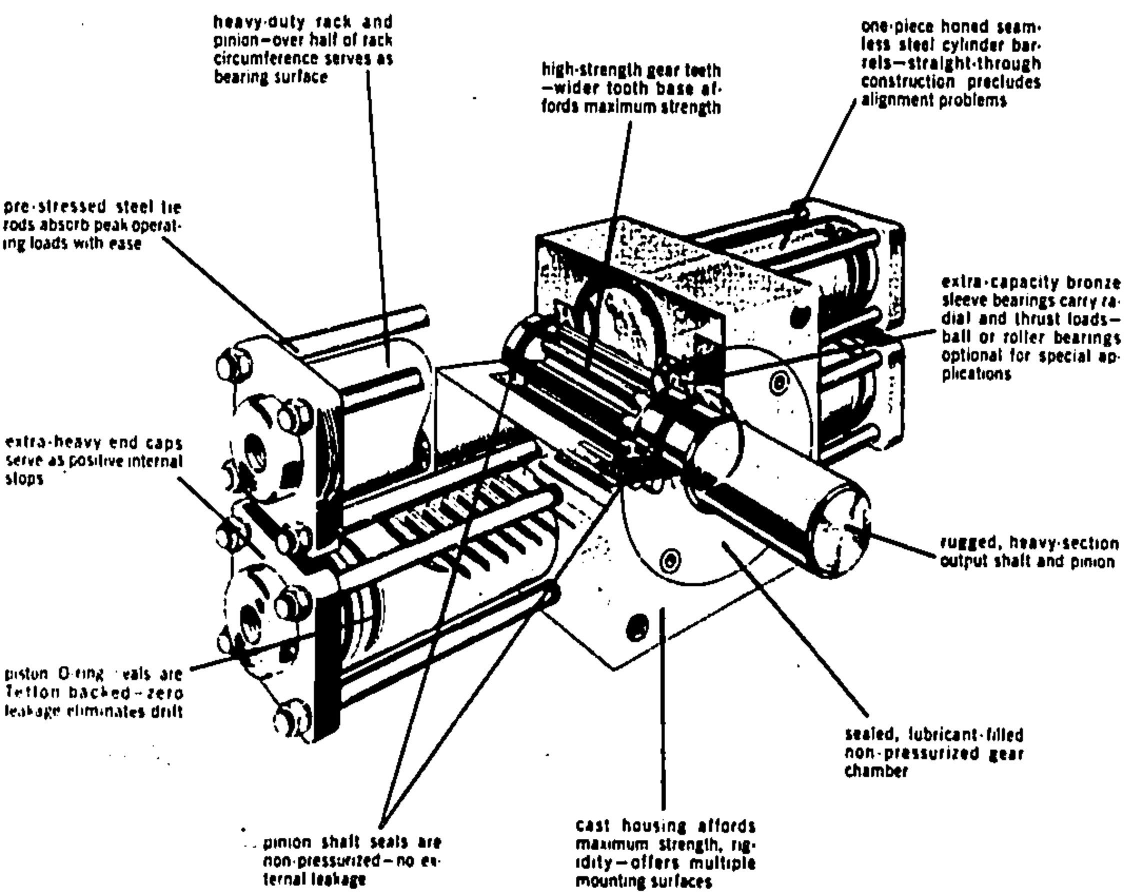



Fig. 17B. A FLO-TORK rotary hydraulic actuator.

4. <u>Check Valve Assembly</u>, $V_1$, $V_2$, $V_3$ and $V_4$

As has been described, four check valves are required in the construction of a flow-work exchanger unit. In order to reduce the make-up energy required in the operation, the pressure loss of the check valves should be low. Various types of check valves are available on the market. Some simple ones are inexpensive but usually have high pressure loss and cause water hammer. More sophisticated valves usually have an expanded cross section at the valve body to reduce pressure loss and have provisions to eliminate water hammer. Check valves are commonly used in hydraulic fields. These valves are, however, not suitable for use, because they are not corrosion resistant to sea water and they have a large pressure loss. Low pressure loss, corrosion resistant valves operable under high pressure (say 2000 psi) are available. Manufacturers of such valves are

Charles Wheatley Co., Tulsa 20, Oklahoma

Mueller Steam Speciality Co., Inc., Brooklyn, N. Y. 11222

Smolensky Valve Co., Inc., Bedford, Ohio 44014

Durable Manufacturing Co., New York 6, N. Y.

The function of the check valve assembly of a flow-work exchanger unit is similar to that of the check valve assembly of the fluid cylinder of a duplex plunger pump. Therefore, a fluid cylinder with its four check valve assembly may be used in the construction of a flow-work exchanger unit. We recommend the use of such a fluid cylinder, since it simplifies the construction greatly.

Hydraulic check valves were used in constructing the two units simply because they were readily available at the time of constructing the units. These should be replaced by corrosion resistant check valves.

5. Control of Valve Actions

It has been stated that a double-plunger four-way valve is used to control the flow in a flow-work exchanger unit where the four-way valve is actuated by an air cylinder, and the air flow to the air cylinder is controlled by a solenoid operated air control valve. Therefore, by controlling the solenoid action, the operation of the unit can be controlled. The solenoid should be actuated whenever the floating pistons in the displacement vessesl reach the respective extreme ends of their strokes. There are several ways to control the solenoid action - one is to use a programming timer, another is to use micro-switches installeo in the displacement vessels and still another is to use a pressure switch. These are described as follows:

a. Programming timers

There are various types of timers available on the market. We have found that a programming cam timer works quite well. Referring to Figures 18A and 18B a programming cam timer has one or more adjustable cams C attached to a cam shaft D which is rotated by a small synchronous motor M through a gear rack G. Cycle action can be adjusted by adjusting the cam and the cylce time can be varied by replacing the gear rack. The timers used in the two flow-work exchanger units can be varied between 10 sec. to 90 sec. Variable speed programming cam timer is also available. Manufacturers of programming cam timers are

Industrial Timer Corp., Parsippany, N. J. 07054
Los Angeles, Calif. 9007

Bristol Motors/Timers, Old Saybrook,Conn.

Automatic Timing and Controls, Inc., King of Prussia, Pen. 19406

Precision Timer Co., Inc., Westbrook, Conn.

Bayside Timers, Inc., Flushing, N. Y. 11358

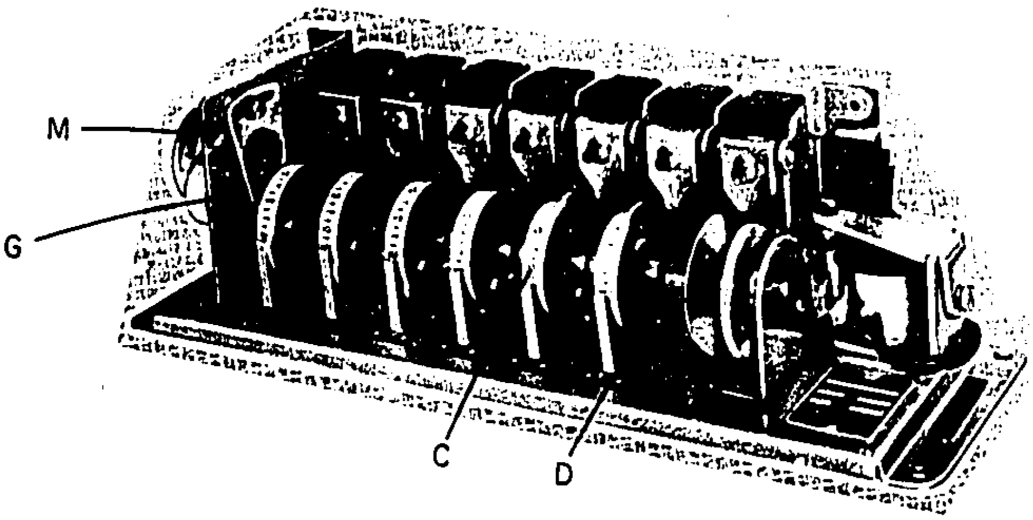



Fig. 18A. Synchronous motor drive programming cam timers, multi-switch type.

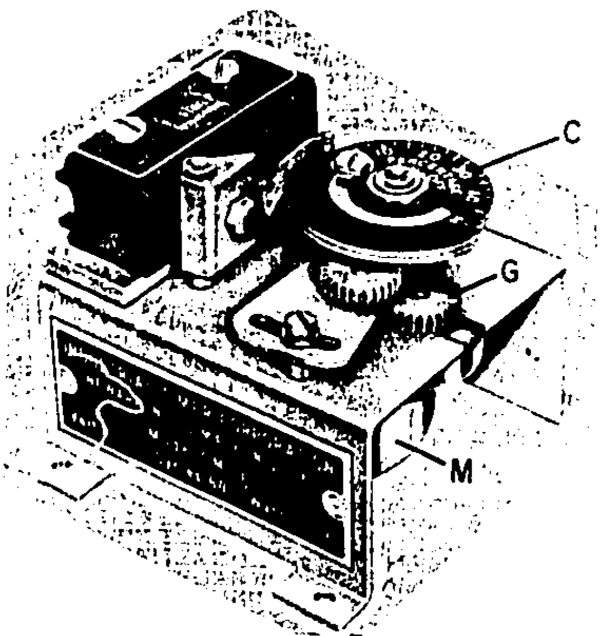



Fig. 18B. Motor driven programming cam timers, single-switch type.

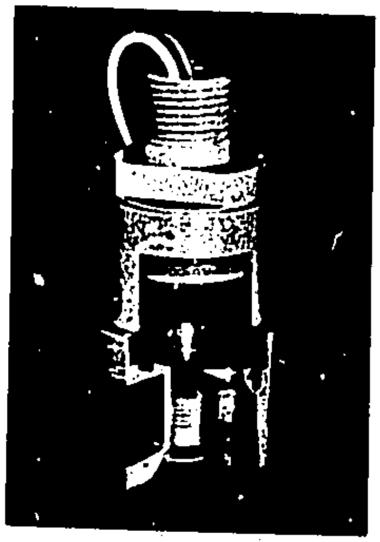


Fig. 18C. Dual snap pressure switch.

AEMCO Division, Didtex Incorp., Mankato, Minn. 56001

Elder Corporation, Palmyra, Wis., 53156

Zenith Controls, Inc., Chicago, Ill. 60610

b. Control by sensing movement of pistons, bladders or diaphragms

Control of the solenoid action can also be accomplished by installing micro-switches within the displacement vessels. Referring to Fig. 2, we may install micro-switches at the a, b ends of the displacement vessels. Whenever either the floating pistons $M_1$ and $M_2$ respectively touch the a end of $O_1$ and the b end of $O_2$, or $M_1$ and $M_2$ respectively touch the b end of $O_1$ and the a end of $O_2$, the solenoid is actuated. Manufacturers of micro-switches are many.

c. Control by either a pressure switch or a differential pressure switch

Referring to Figs. 6 and 12, whenever the floating pistons $M_1$ and $M_2$ come to the end of their strokes, the flows in the lines $L_1$, $L_2$, $L_3$ and $L_4$ stop. Therefore, the pressures in the lines increase and the pressure differentials across the valves diminish. By installing either a pressure switch in a line or installing a differential pressure switch across a pipe fitting or across a valve, the piston movement can be sensed. By connecting the pressure switch or the differential pressure switch to the solenoid, the fluid flow in a flow-work exchanger unit can be controlled. Figure 18C shows such a pressure switch. Manufacturers of pressure and differential pressure switches are

Custom Component Switches, Inc., Chatsworth, Calif.

Consolidated Control Corp., Bethel, Conn.

Meletron Corporation, Los Angeles, Calif.

BEC Pressure Controls, Co., Davenport, Iowa.

## VIII. SUGGESTIONS FOR THE CONSTRUCTION OF FUTURE UNITS

Based on the experience gained during this contract work, the following suggestions are made for the construction of future units:

(1) When floating piston type vessels are used, provide a small hole in each floating piston to fascilitate venting of air entrapped beneath the piston. This hole may be covered by a solenoid operated cover which is normally closed and is opened only during venting operation. This simplifies the start-up operation greatly.

(2) When bladder type vessels are used, the normal holding capacity of each of the bags, the capacity of the bag without stretching should be equal to the volume of each outer shell. This enables smoother fluid flow in the unit.

(3) The check valves used should be corrosion resistant and have low pressure losses. We may also use a fluid body of a double acting plunger pump with its check valve assembly.

(4) The high pressure circulation pump should have extra developed head to overcome pressure drop which arises in the reverse osmosis unit to be installed.

(5) An accumulator must be installed in high pressure lines to prevent water hammer.

(6) An automatic control must be provided to synchronize the movements of the floating pistons or the bladders, $M_1$ and $M_2$. This may be achieved by installing micro-switches within the displacement vessels. If the two pistons do not arrive at their respective ends within an allowed time limit flows are automatically regulated to achieve this.

Figure 19 shows a schematic illustration of a suggested unit. It uses a fluid-body of a duplex plunger pump as its check valve assembly. Air vents are provided on the floating pistons and an accumulator is provided on line $L_5$ to eliminate line shock caused by water hammer.

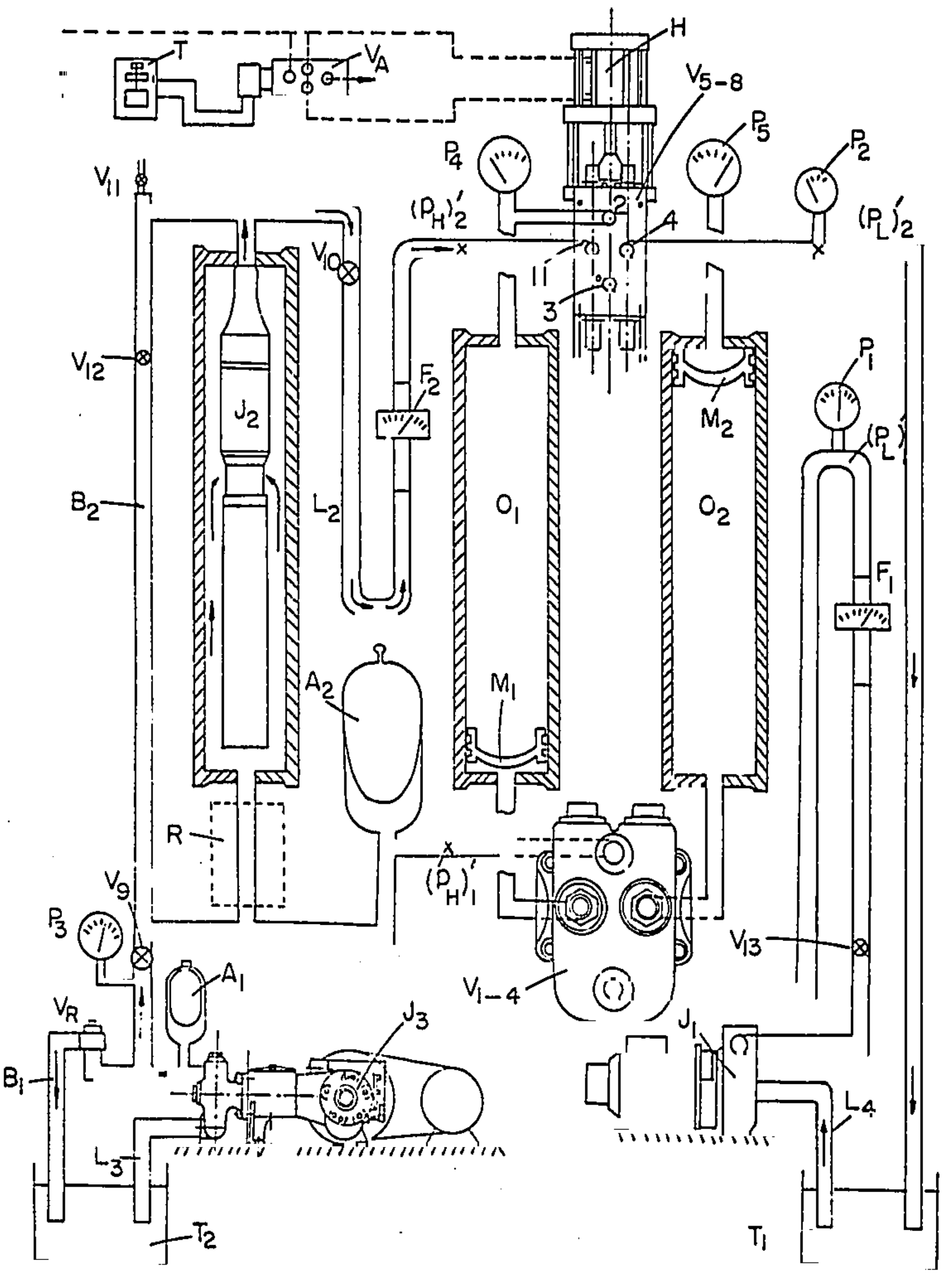



Fig. 19. Schematic illustration of a future unit.

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## Appendix

Recently, the Warren Rupp Company of Mansfield, Ohio has started manufacturing a new type pump under the tradename "Dynaflex." The pump combines the smoothness and dependability of centrifugal pumps with the abrasive and solids handling capabilities of diaphragm pumps, together with the high head potentiality of various positive displacement pumps. The construction of the pump has certain similarities with that of a flow-work exchanger. By analyzing the similarities and the differences between them, we can obtain a better understanding of the technical and economical feasibilities of the flow-work exchanger. Therefore, a brief description of the "Dynaflex" pump is given below:

Referring to Figs. 20-A, 20-B, the heart of the "Dynaflex" is a large free diaphragm (1) clamped between two symmetrical pumping chambers. Water is alternately pumped in and out of the upper chamber (2) by a standard centrifugal pump (3). The diaphragm moves in response to the fluid volume in the upper chamber, reacting on the fluid in the lower chamber (4), creating a pumping action. The diaphragm serves merely as a separating "membrane" between the two liquids and is therefore not subjected to stress. Inlet and outlet check valves (5) control movement of pumped material. A diaphragm sensing device (6), monitors the movement of the diaphragm and controls the shifting of the four way valve (7). As the diaphragm approaches its low position, the sensing device shifts the four way valve to the suction stroke. As the diaphragm approaches the top position the sensing device signals a shift to the discharge stroke. The four way valve diverts the flow of water from the drive pump in a completely "shock free" transition, creating suction and discharge stroke with no water hammer.

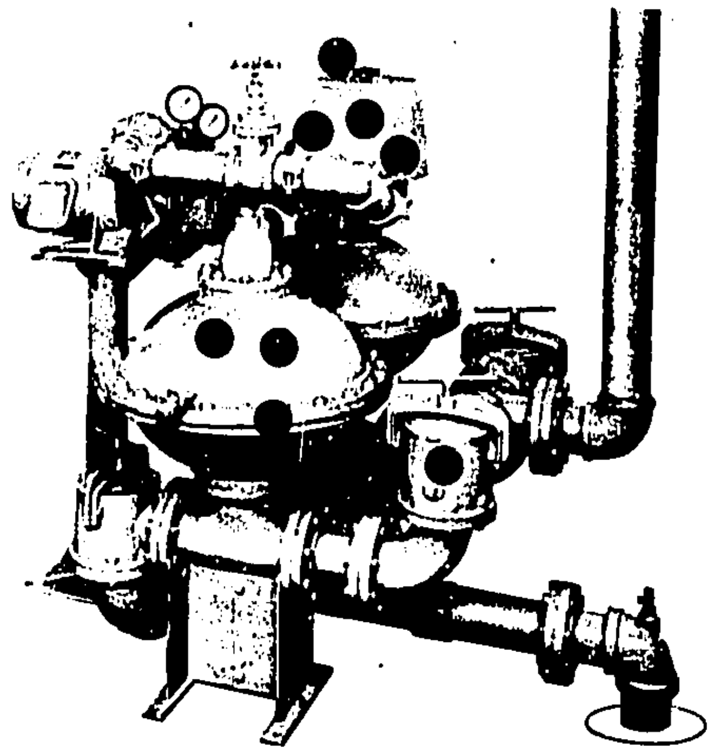

Fig. 20A. A duplex DYNAFLEX pump.

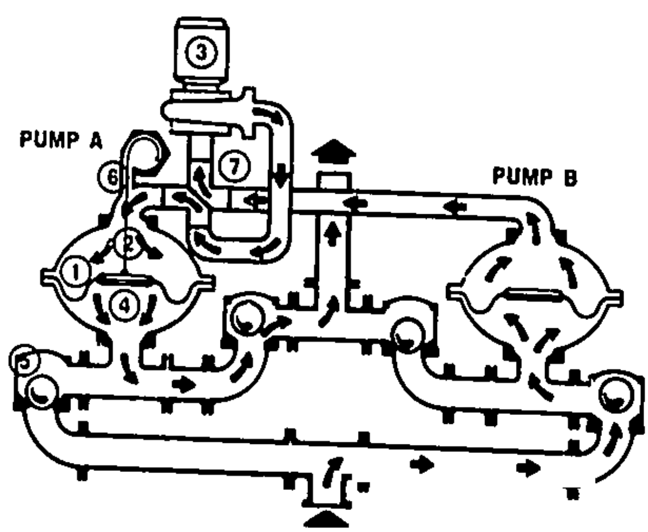


Fig. 20B. The diagrammatic drawings showing the duplexed unit.

The components used in the unit are described as follows:

(1) THE DIAPHRAGM CHAMBERS - The diaphragm chambers are symmetrical with sloping walls to prevent "Hang Up" and to afford smooth flow of foreign matter from the lower chamber. The capacity is fifteen gallons.

(2) THE DIAPHRAGM - The diaphragm is a separating membrane between two fluids and is not subjected to stress, giving it virtually unlimited wear life. Pumps developing hundreds of pounds per square inch are possible.

(3) DIAPHRAGM POSITION SENSING DEVICE - The diaphragm position sensing device monitors the movement of the diaphragm and signals the shifting of the four way valve controlling the direction of flow.

(4) THE FOUR WAY VALVE - This was developed specifically for the "Dynaflex". The four way valve is symmetrical with a ball bearing mounted "butterfly" and full flow passages throughout and provides gradual shifting resulting in shock-free pumping.

(5) THE CONTROL MEANS - A pneumatic system is used and by simple needle valve adjustments the shifting of the four way valve is infinitely variable. (Only 2 c.f.m. required)

(6) THE SHIFTING CYLINDER - This is another first for the "Dynaflex." A differential piston type cylinder, with a two to one ratio provides equal shifting force in either direction. Air over oil is used on the small side for a viscous dampening effect and lubrication of the piston. No stops are needed to limit travel.

(7) FLEXIBILITY OF ADJUSTMENTS - Needle valve adjustments control the shifting cylinder and four way valve, while independently adjustable cams on the position sensing device control the length of stroke of the diaphragm.

(8) WATER SUPPLY - A water storage tank replaces this segment of the "Dynaflex" unit on the simplex installation. The tank is large enough to provide an air cushion and dissipation of heat build-up.

(9) THE DRIVE PUMP - An efficient centrifugal pump with good suction (NPSH) characteristics is used. Pumping clean water, its life is virtually unlimited.

(10) THF CHECK VALVES - Suction and discharge check valves are identical. Easily replaceable valve seats, "T" handle clamp cover and drain plugs provide for quick and easy access, cleaning, or maintenance. Standard ball checks pass solids up to one half the size of the piping. Optional swing check valves allow up to full pipe size solids pumping capability.

(11) COOLING UNIT - A fan type cooling unit is furnished, when necessary, on the Duplex unit to dissipate heat.

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